A study of the wall vibrations excited during the playing of lip-reed instruments

Thesis

Link(s) to article on publisher's website:
http://dx.doi.org/doi:10.21954/ou.ro.0000adcd

A STUDY OF THE WALL VIBRATIONS EXCITED DURING THE PLAYING OF LIP-REED INSTRUMENTS

James Whitehouse

Thesis submitted to the Open University in part fulfilment of the requirements of the degree of Doctor of Philosophy

Technology Faculty

The Open University

31st. October 2003

DECLARATION

I declare that this thesis has been composed by me and that the work is my own.

ABSTRACT

The question of whether vibrations induced in the wall of a played instrument affect the tone produced has been the subject of numerous studies. While instrument manufacturers and musicians regularly claim the ability to distinguish between wind instruments of different materials, researchers today still debate the legitimacy of this claim and the mechanism by which it could occur.

In this thesis, a series of experiments is presented in which a simplified wind instrument, consisting of a trombone mouthpiece coupled to a section of brass pipe, is blown using an artificial mouth and the induced wall vibrations measured using a Laser Doppler Vibrometer. The brass pipe's natural resonance frequencies and bending mode shapes are established and compared with the velocity amplitude variation along the pipe induced by an artificially blown note. The induced wall vibrations are shown to occur at the harmonics of the played note and to match the shapes of the bending modes of the pipe at those frequencies.

Adapting this technique, the excitation mechanism by which the induced wall vibrations are set in motion is investigated by decoupling the section of brass pipe, in turn, from the air column and the mouthpiece. Results indicate that it is the motion of the lips against the mouthpiece, rather than air column pressure changes within the pipe, that is the dominant mechanism in exciting wall resonances.

Five pipes of identical dimensions but manufactured from different metals and three pipes of the same brass alloy but drawn to a differing wall thickness are then tested in a series of comparison measurements. The pipes are measured under identical conditions and their structural and induced vibrational velocity measurements compared. Results reveal that the magnitude of the induced wall vibration depends on the material from which the instrument is made and its wall thickness.

PUBLICATIONS ARISING

Conference Papers

Whitehouse JW, Sharp DB and Harrop ND. *The use of laser doppler velocimetry in the measurement of artificially induced wall vibrations in wind instruments.* Proceedings of the International Symposium on Musical Acoustics, Perugia, Italy, September 10-14, 2001; 411 - 414.

Whitehouse JW, Sharp DB and Harrop ND. *The use of laser doppler velocimetry in the measurement of artificially induced wall vibrations in a wind instrument.* Institute of Acoustics, Spring Conference 2002. Past, Present and Future Acoustics and EPSRC Theme Day in Acoustics, Salford, England, March 25-27, 2002; CD-ROM Vol. 2 Part 2.

Whitehouse JW, Sharp DB and Harrop ND. *An investigation into wall vibrations induced in wind instruments constructed from different metals.* Proceedings of the International Symposium on Musical Acoustics, Mexico City, Mexico 9-13 December 2002; CD-ROM.

Invited Publications

Whitehouse JW, Sharp DB and Hill TJW. *Wall vibrations in musical wind instruments.* Acoustics Bulletin (Institute of Acoustics), Technical Contribution, Vol. 28 No.4 July/August, 2003.

ACKNOWLEDGEMENTS

First and foremost, I would like to thank my supervisor Dr. David Sharp. I could go on about the help, support etc. that he has given me throughout my time at the O.U. but all I really need say is that I was extremely fortunate in having a boss such as he.

Thanks go to Dr. Rod Barratt for allowing me to use the facilities at the Department of Environmental and Mechanical Engineering, the EPSRC for funding me, and all staff (academic, secretarial and technical) and students in the Acoustic Research Group.

I would also like to thank Roger Traynor of Lambda Photometrics Ltd. for the kind use of, and help with, a scanning LDV system used when the department was in a transitional stage between systems. His advice with the Polytec system has been much appreciated.

The other big names I would like to express my gratitude to are Dr. Nick Harrop (the one that got away) a great friend who has always sorted me out, Eva Vass for her constant encouragement, Dr. Toby Hill who has helped me enormously in my run up to submission, and Peter Seabrook for all the support he has given me both in and out of the lab. It has genuinely been a pleasure working here.

Finally, all the people I met at the acoustics group at Edinburgh University and at the ISMA conferences. A friendly, close knit community who helped someone new to the subject ‘settle in’.

CONTENTS

LIST OF TABLES

LIST OF FIGURES

Chapter 1

Do Wall Vibrations Influence Sound Production?

1.1 General Introduction

Although the most significant resonances in a wind instrument are associated with the air column, the question of whether the structural resonances have an effect on the tonal quality of the instrument remains a subject of debate. Many instrument manufacturers, musicians and researchers have at one time or another claimed that wall vibrations are important and that it is possible to distinguish between sounds produced by instruments manufactured from different materials. For example, Morral [1986] states "Clearly, the material with which an instrument is made may affect the sound it produces…..Silver gives a trumpet a dark sound because the metal damps high frequency vibrations. Brass, on the other hand, responds to these high frequencies and gives a brighter sound". Similarly, Cocchi and Tronchin [1998] state that "The influence of material and obsolescence..….is well known amongst musicians. It is well known that a gold flute sounds better than an alloy flute, though it is not well established where the difference lies". These claims are backed up by wind instrument players who suggest that alternative materials or finishes should be used to attain enhanced musical qualities and by brass instrument makers, who advertise a range of materials for their products believing that the materials impart differing musical qualities.

Despite all these claims, systematic experimental studies have so far failed to demonstrate conclusively that it is possible to distinguish between the sounds produced by instruments of different materials. This is acknowledged by Gautier and Tahani [1998] who state "The question of the effect of wall vibrations on the tone of a musical wind instrument remains open: we still do not know if this phenomenon is important for the emitted tone or not".

1.2 Possible Wall Influences on the Produced Sound

There are a number of different ways by which wall vibrations might affect the sound produced by a brass instrument (Figure 1-1). The movement of the walls could radiate sound directly to the surrounding environment. The frequency of this sound would be related to the frequency of movement and the sound would emanate most strongly from areas of maximum movement. Alternatively the walls could interact with standing waves in the air column. This would affect the air column vibrations within the pipe and thus change the harmonic structure of the radiated sound and alter its tone. Finally, the wall vibrations could be transmitted through the mouthpiece to the player's lips and may then cause the player to adjust their embouchure in response.

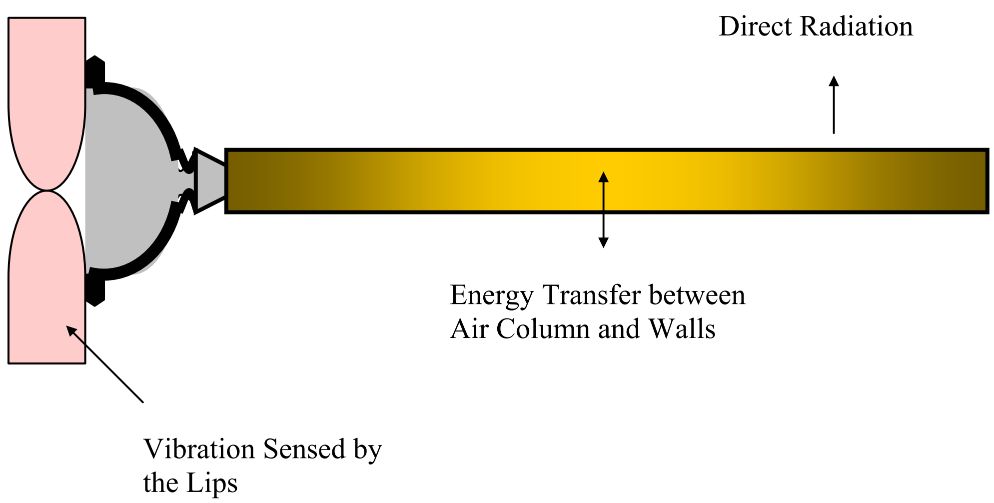


Figure 1-1. Possible ways in which an instrument's wall vibrations may influence sound

Whatever the mechanism, the wall vibrations must be reasonably large to make a significant contribution to the radiated sound [Fletcher and Rossing 1991]. So, any change in the sound is more likely to occur when natural wall resonances are being excited and the vibrations are of maximum amplitude. If resonant at certain frequencies, for example those of a musical scale, the effect could be constructive, detrimental or insignificant. Pyle [1981] has shown that there is a “not insignificant” difference between the acoustic radiation at high frequencies of lacquered and plated instruments.

The strengths and frequencies of the structural resonances of a lip-reed instrument depend mainly on the wall thickness (the thicker the wall, the lower the vibrational amplitude), the pipe geometry (the length and radius of the pipe) and the wall properties (type of material, alloy, finish).

In the next section, a review of the literature concerned with investigating the effect of the wall thickness, geometry and material on the wall vibrations and the sound produced by wind instruments is presented.

1.3 Literature Review

Since as early as the nineteenth century, researchers have been investigating the question of whether the material of manufacture has an effect on the tone quality of wind instruments. In his paper of 1909, Miller summarised this early work and also carried out further experiments using organ pipes designed so that their walls could either vibrate freely or the vibrations could be suppressed. His findings indicated that the wall vibrations did affect the sound produced by the organ pipes.

Miller's claims were backed up by Lottermoser [1938] who studied the sound spectra produced by organ pipes constructed from different materials, concluding that measured variations were a result of this material change. However, Boner and Newman [1940] also studied the sound spectra of pipes constructed from different materials. They concluded that the material change had negligible effect on the generation and emission of sound from the mouth and top of the flue. They also claimed that any emission from the walls was small in comparison with that from the mouth and flue top. This same conclusion was drawn for brass instruments when Knauss and Yeager [1941] mechanically oscillated the walls of a cornet at different playing frequencies and found negligible sound radiation. They concluded that any contribution by the wall is completely masked by the air column vibration. However, Lawson and Lawson [1985] inferred from Martin's studies [1942] that the factor responsible for the asymmetric shape of the sound field radiated from an axially symmetric bell was the bell wall vibration.

An investigation by Backus [1964] used an artificial embouchure to blow a clarinet which had sound absorbent material inserted to absorb the sound radiated from the end of the air column. The sound radiated from the body of the instrument was measured and found to be 48 dB below that measured when the absorbent material was removed. In the same paper, Backus also compared tubes of different materials blown on the artificial embouchure, discovering that the frequencies and Q factors of the resonance modes of the air column were independent of the material. His overall conclusions were that the wall vibrations of a woodwind instrument do not affect the tone of the instrument, either by radiating sound directly or by affecting the harmonic structure of the internal standing wave.

Later Backus and Hundley [1966] investigated the effect of wall vibrations on the tone of flue organ pipes both experimentally and theoretically. Backus [1965] had previously found that in cylindrical organ pipes, unlike rectangular pipes, wall material negligibly affects tone colour. In the 1966 paper, the cause of this difference was shown to be the degree by which the instrument wall yields to the internal pressure of the standing wave. Backus and Hundley claimed that for a rectangular bore instrument, the wall yield and sound radiation are measurable whereas they are negligible for an accurate cylindrical bore instrument. In addition, the paper also presented a simplified model of the acoustic resonance frequency shifts of an air column caused by wall vibrations. This model revealed that, even when the wall vibrations are measurable, they do not affect the internal standing waves significantly. Backus and Hundley therefore concluded that wall vibrations do not affect the steady-state vibrations of organ pipes.

The effect of material on flute tone quality was examined by Coltman [1971] through psychoacoustical tests. Three keyless flutes, of identical internal dimensions but constructed from different materials, were played, out of sight, to musically experienced observers. With the removal of any visual or tactile clues, the flautists were unable to distinguish between the three materials. In the same vein, there was found to be no statistically significant correlation between the listeners' scores and the instrument material.

Research into the vibrations of the walls of wind instruments has not been solely concentrated on organ pipes and woodwind instruments. Work has also been carried out on brass instruments, in many cases concentrating on the bells of such instruments. In 1979, Wogram [1979] using trombones of different alloys reported a 3 dB difference at 3-5 kHz between those of brass and nickel-silver. Similarly, Pyle [1981] using French horns with lacquered, silver-plated, or uncoated bell flares found a slight reduction of relative sound pressure levels at high frequencies for the lacquered bell only. A later paper by the same author [1998] uses a trombone where bells of different alloys can be used interchangeably. The author reports noticeably different spectra when lower pitched notes were played using different bells, although when higher pitched notes were played the spectra were less different. All these findings are backed up by the work of Lawson and Lawson [1985] who recorded sound level measurements on several annealed French horn bell flares of varying material and hardness. The sound level of the unannealed brass flares was higher in the 1-3 kHz range than the annealed, but the opposite was observed for the nickel silver. Indeed, Lawson and Lawson state that their measurements support musicians' claims that the material of a French horn affects tone quality.

Although the experimental work carried out by Wogram, Pyle, and Lawson and Lawson appears to indicate that the wall vibrations in the bells of brass instruments have a significant effect on the tone quality of the sound produced, theoretical studies by Watkinson and Bowsher suggest that this shouldn't be the case. In their paper [1982], they used finite element techniques to model the frequencies of the modes of vibration of trombone bells, finding that all the frequencies lay within the full range of played frequencies. They deduced that a variation in the wall material affected the responding mode frequencies but had little effect on the overall level of response.

Richard Smith, a scientific researcher and manufacturer of brass instruments, performed measurements using six brass bells of varying thickness which were in turn coupled to a trombone body [1981, 1986]. Holographic interferometry confirmed that the amplitude of vibration of the bells was dependent on the material thickness. Using a siren (originally developed by Wogram [1979]) to drive the trombone, Smith found that the bell thickness had a significant effect on the sound spectra measured at the player's ear position, due to sound directly radiated from the bell itself. However, psychoacoustical tests revealed that players were still unable to distinguish between the six bells.

In the 1990s, researchers began to employ the measurement technique of laser Doppler vibrometry to aid the study of wall vibrations. Zipser and Franke [1996, 1997] used scanning laser vibrometry to measure both the structural eigenmodes of a vertically hanging thin-walled cylindrical tube (chosen to have similar dimensions to an organ pipe) and the wall vibrations induced when the air column was excited into vibration by a loudspeaker. They noted that, in the latter case, the wall vibration patterns were not images of the air column pressure fluctuations and concluded that the wall and air column represented a complex coupled vibrating system. Both damped and undamped pipes were measured to estimate the influence of the body vibrations on the sound emission spectra with the result being that no significant differences were found between the spectra.

Similar work was carried out by Runnemalm, alone [1997] and in collaboration with Zipser and Franke [1999]. In these papers, Runnemalm takes the view that there certainly must be differences between the sounds produced by organ pipes manufactured from different materials and using different tooling methods, because well-trained musicians claim to be able to notice them. She reasons that the research to date has simply failed to explain the origin of the differences. Runnemalm attempted to provide experimental evidence for the said behavioural differences by using TV holography and laser Doppler vibrometry to measure the free structural eigenmodes of eighteen organ pipes and the structural responses when their air columns were forced to vibrate at the harmonic partials. Measurements of the wall vibrations induced when the pipes were blown were also made. From all these measurements, Runnemalm concluded that the structural vibrations were strongly influenced by pipe material and tooling. However, her measurements of sound intensity did not reveal with certainty any corresponding differences in the sounds produced by the organ pipes.

Angster et al [1998] reached similar conclusions to those of Runnemalm through objective measurements made on rectangular wooden, and cylindrical metal, pipes which had been ‘specially tuned’ by an organ builder to improve their timbre. This tuning involved removing wall material from the pipes; i.e. thinning sections of the walls. Although laser Doppler vibrometry measurements revealed that the tuning resulted in large increases in vibration amplitude, only slight differences were observed in the sound signal before and after the wall material was removed.

Malte Kob [2000], through recordings made when a metal organ pipe was blown both with and without damping applied to the walls, confirmed the observations of Angster and Runnemalm. He noted that although some structural modes were excited when the pipe was blown, the wall vibrations didn’t have any noticeable effect on the stationary sound spectrum. However, Kob also noted that the wall vibrations did significantly change components in the transient spectra which may explain any perceived differences in timbre. This is a point considered by Coltman back in 1971 when he questioned the pertinence of using steady tone measurement in the study of the influence of wall vibrations on timbre, noting that the harmonic content of the stationary spectrum is alone not sufficient to characterise an instrument.

Whilst Kob gave some consideration to the question of by what mechanism the wall vibrations perturbed the transient spectra, a more in-depth analysis was provided by Gautier and Tahani [1998]. Using a simplified model of a clarinet, the effects of coupling between the walls and the internal air column and between the walls and the surrounding environment are investigated. Shifts in the instrument's resonance frequencies due to the couplings are calculated and related to small tone changes.

The debate has not subsided as we enter into the new millennium with Widholm et al [2001] performing psychoacoustical tests using flutes constructed from different materials. Spectral analysis revealed big differences in the sound level and colour of tones caused by the player, and just measurable (but not perceivable) sound colour differences caused by the material. From their statistical analysis they concluded that there was no evidence that wall material has any appreciable effect either on sound colour or on the instrument's dynamic range.

Even more recently, Moore et al [2003] compared the acoustic spectrum of a damped trumpet bell to one freely vibrating. They showed that the reduction in the amplitude of vibration due to damping caused a change in the spectrum, the relative power in some of the harmonics changing by a factor of two. They claim the results are in agreement with anecdotal evidence from brass players that damping the bell vibration increases the relative power of the lower harmonics. The results are explained by assuming that the bell vibrations cause a change in the viscous boundary layer within the bell.

Though the subject of numerous studies over many years, the question of whether vibration in the wall of an instrument affects tone remains an open one. While most musicians and manufacturers believe the thickness and wall material of their instrument affects the sound, to scientists it is still a topic of debate.

1.4 Experimental Difficulties in Measuring Variations in Tone Quality

Over the years, there are a number of factors which have been problematic when comparing the wall vibrations of wind instruments and their possible influence on the sound produced:

- When the excitation mechanism for the instrument is an actual musician a number of problems can occur. A musician would find it difficult to maintain a constant pressure over the time necessary to complete a measurement. It is highly improbable that a specific note, with its attendant harmonics, could be reproduced repeatedly allowing a series of measurements to be made for comparison. Smith [1986] found that, when using musicians to measure acoustic differences between bells, "the variability between different tests was greater than the differences between instruments". Players may also subconsciously adjust to the type of material from which the instrument is made. It would be quite natural for a musician to believe, for instance, that gold or platinum flutes play with a superior tone to that constructed from a basic alloy and so play to their maximum potential. There is also a question of differential 'warming up' rates for instruments constructed from different materials by the players breath.

- Instruments constructed from more expensive materials will have had much more care taken over their tooling in the manufacturing process. This could be a direct cause of a change in tone quality.

- Small changes in a pipe's geometry can cause significant changes in its sound, making comparisons between pipes difficult. It can be very difficult to construct two instruments with identical internal dimensions.

- People perceive sound differently from each other. Kob [2000] states that a human's perception is more sensitive but much more complicated than measuring equipment. A specific property of sound distinguished by a listener is not necessarily measurable, and a measured one not necessarily audible.

- Instrument makers and good musicians are very experienced in listening to subtle aspects of the sound, which most people would overlook. Also some people may just have keener discriminatory senses.

- As the phenomena involved are complex, in theoretical studies approximate methods are used and it is difficult drawing conclusions from simplified models.

This thesis is concerned with the investigation of the mechanism by which vibrations are excited in the walls of brass instruments and their effect, if any, on the sound produced. Consequently, of the problematic factors outlined previously, the main ones that needed to be addressed were the blowing mechanism and the consistency of the different measured instrument's dimensions and tooling.

To overcome the problem in the variability introduced by using a human player to blow the instrument, an artificial mechanism was used. To this end a blueprint for an artificial mouth was obtained from the Musical Acoustics Research Group at Edinburgh University and constructed within the Acoustics Research Group of the Departmental of Environmental and Mechanical Engineering at the Open University. An artificial lip system offers great benefits in long-term stability, ease of instrumentation and the ability to make small and reproducible adjustments to the embouchure.

The problem of variability introduced by differences in the dimensions of the instruments was overcome by using straight sections of cylindrical piping as the instrument bore. In their [1997] study, Zipser and Franke believed that to help understand the complex behaviour of organ pipes it would be helpful to first investigate vibrating thin walled cylindrical tubes. In this study, by adding a mouthpiece to simple lengths of pipe, wind instruments constructed from different materials but identical dimensions are created. These can be attached to the artificial mouth in turn and blown at a specific note, recording any vibrational change along with any concurring alteration in tone.

Due to the size of most instruments, and the limitations of space within the anechoic chamber where the measurements were to take place, a length of not more than one metre was decided upon. The internal diameter of the pipes was pre-set, as the tubes will have to accommodate a trumpet/trombone mouthpiece for blowing purposes.

1.5 Aim and Contents of Thesis

Aim

1. To determine the main excitation mechanism inducing the structural vibrations in the wall of a brass wind instrument.

2. To determine the effect of both wall material and wall thickness on the vibrational response of a simple lip-reed wind instrument when artificially blown and to gain an insight into their possible influence on the tone produced.

Contents

Chapter 2 describes the theory behind the operation of a brass wind instrument. The basics of sound generation are described, as is how this sound is affected by instrument geometry. This gives an understanding of how the simple wind instrument, used throughout the study, functions.

Chapter 3 describes Laser Doppler Vibrometry (LDV), a non-invasive optical measurement technique used to make high resolution measurements of vibrational velocity variations at specific points on the surface of a test object.

In Chapter 4, the dynamics of cylindrical shells are reviewed and methods of exciting a cylindrical test object investigated. The suitability of the LDV test system is assessed by exciting a section of plastic pipe to determine its structural resonance frequencies. These experimentally derived frequencies are compared to theoretical predictions calculated from the pipe's physical parameters.

Chapter 5 describes a series of experiments designed to study the wall vibrations induced when a simple brass instrument is artificially blown. The simple instrument consists of a cylindrical acoustic resonator (metal pipe) and a trombone mouthpiece permanently fixed to the front plate of an artificial mouth. The structural resonances are ascertained as they were for the plastic pipe. These are then compared to the velocity amplitude variation along the pipe wall when the instrument is blown.

Chapter 6 investigates the excitation mechanism inducing the vibrations in the wall of the simple wind instrument seen in Chapter 5 in an attempt to divine the primary process by which they are generated. This is achieved by comparing the results from two experiments. Firstly, the pipe is decoupled from the mouthpiece while the standing wave (note) is maintained by the insertion of a short flexible tube. Secondly, a section of aluminium tube is inserted into the instrument and fixed to the mouthpiece, effectively decoupling the air column from the instrument wall.

Using the experimental system developed over the course of the study, Chapter 7 compares the structural resonances with the induced wall vibrations when artificially blown for a simple wind instrument where the wall material and wall thickness has been altered. To do this, five pipes of equal dimension but constructed of different material (brass, copper, aluminium, stainless steel and titanium) are fitted in turn to the mouthpiece while it is sounding a steady note. A second set of measurements was made for three brass pipes of equal internal dimensions but varying wall thickness.

Chapter 8 summarises the results from the study with respect to the initial aims and discusses proposals for further research.

Chapter 2

How Does a Brass Wind Instrument Work?

2.1 Introduction

Sound production in a brass wind instrument is initiated by a musician blowing a stream of air into the instrument whilst 'buzzing' their lips. This excitation causes the air column contained within the instrument to vibrate at specific frequencies resulting in the production of a musical note. In addition to these air column vibrations, blowing a brass wind instrument also sets up vibrations in the walls of the instrument. The remaining chapters of this thesis are devoted to the investigation of the origin of these wall vibrations and their possible influence on the sound produced by the instrument. However, it is the vibration of the air column that is the dominant factor in determining the pitch and timbre of the note produced by a brass wind instrument. In this chapter, an overview of basic air column theory is given, describing why a given air column tends to vibrate at specific frequencies, how the geometry of the air column is responsible for determining these frequencies, and how the lips can be used to excite the air column vibrations.

2.2 Simple Air Column Theory

The air column enclosed within a wind instrument is a resonant system. As such, there are certain frequencies, called natural frequencies, at which it will tend to vibrate. Resonance occurs when the frequencies of an applied excitation source match the natural frequencies of the air column causing standing waves to be set up. Each of the natural frequencies at which an air column vibrates is associated with a particular standing wave pattern.

2.2.1 Creation of a Standing Wave

Standing waves in an air column are actually a composite of two or more travelling longitudinal waves.

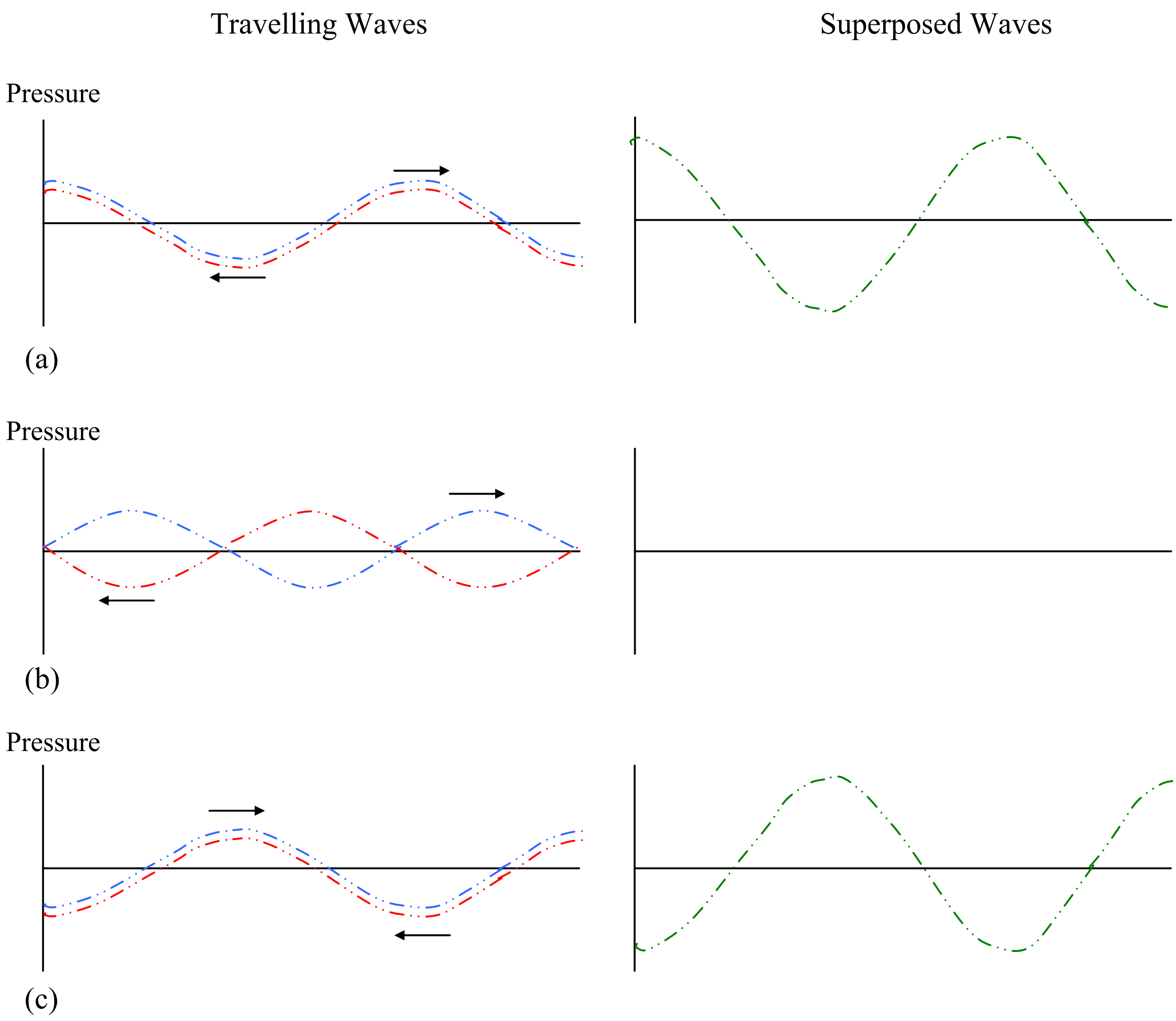


Figure 2-1. Formation of a standing wave from two travelling waves

These travelling waves consist of regions of high pressure (compressions) and low pressure (rarefactions) formed by neighbouring air molecules pushing and pulling against each other after being subjected to some vibrational excitation.

Consider an infinitely long section of tube with two sinusoidal pressure waves of equal wavelength, frequency and amplitude moving towards each other from opposite directions. As the two waves pass, their peaks and troughs move in and out of phase with each other. When the waves are in phase (see Figure 2-1(a)), the high pressure peaks (and the low pressure troughs) of the left-travelling wave overlap the high pressure peaks (and the low pressure troughs) of the right-travelling wave. The two waves superpose resulting in peaks and troughs of twice the amplitude of the individual waves. After each wave has moved on one quarter of a wavelength so that the two waves are exactly 180° out of phase (see Figure 2-1(b)), the high pressure peaks of the left-travelling wave overlap the low pressure troughs of the right-travelling wave. Again, the two waves superpose with the net result being that they now cancel each other out. One quarter of a wavelength further on and the two waves are once again in phase (see Figure 2-1(c)). As the two waves continue to advance, this cycle repeats. The resultant disturbance is known as a standing wave (Figure 2-2). The solid curve shows the maximum displacement in one direction and the dashed curve shows the displacement one half-cycle later.

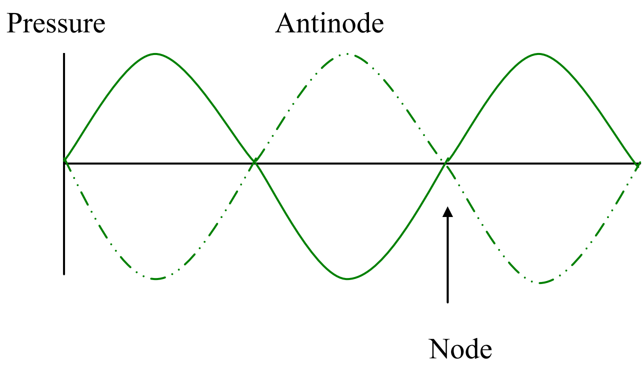


Figure 2-2. Pressure standing wave.

Standing waves are characterised by locations where the pressure amplitude is always zero. Such positions are called nodes and are points where an increase in pressure in the first wave is always matched by a decrease in pressure in the second wave. The distance between any two nodes is half a wavelength. Halfway between the nodes are locations called antinodes. These are positions where the amplitude of the pressure fluctuation is a maximum.

When considering standing waves in a musical instrument, it is important to note that the excitation source is at one end of the air column only. Therefore, the production of a standing wave within the bore must involve internal reflection of the compressions and rarefactions. The natural frequencies at which standing waves occur are determined by the length of the air column and the type of reflection that takes place at the two end boundaries.

2.2.2 Air Column Resonance - Open Ended Tubes

To more easily understand how standing waves are set up in an instrument we first need to consider a simple cylindrical tube with boundary conditions of being open at both ends.

If a rarefaction (region of low pressure) is introduced into one end of the tube, surrounding air will want to move into this region to restore normal pressure, some coming from inside the tube. Within this constrained environment the movement of air creates a new low pressure region farther down the tube, a process which continues until the rarefaction reaches the opposite end. To maintain normal pressure surrounding air particles from the exterior environment once again move into the low pressure region. Through their inertia they continue moving a small distance into the tube creating, due to the constraint of the tube walls, a compression (high pressure region) inside the end of the tube. The pressure wave has undergone reflection due to the discontinuity (change in environment) at the end of the tube and has been inverted from a rarefaction to a compression. This weaker compression propagates back along the tube till it is once again reflected at the discontinuity at the opposite end of the pipe and inverted from a compression back to a rarefaction. Only partial reflection actually occurs at the ends with the rest of the sound energy transmitted into the surrounding environment. Due to loss into the surrounding environment and friction (viscous loss) with the tube wall the process quickly breaks down.

For resonance to occur and a standing wave to be set up, compressions and rarefactions must be periodically introduced at one end of the tube. Taking the previous example, when the returning compression reached the input end of the tube, it underwent partial reflection and was also inverted to produce a weaker rarefaction. If another rarefaction were to be introduced at this point, it would act to reinforce this weaker rarefaction. If this happened on a regular basis, a standing wave would be set up and maintained.

Standing waves can be described in terms of the varying pressure. For an open ended tube, the pressure at each end is always atmospheric so the two points must be pressure nodes. As mentioned previously, the distance between adjacent nodes is half a wavelength so the length of the air column L must be equal to an integer number of half wavelengths.

$$L = n\,\lambda/2 \tag{2.1}$$

n = An integer number (n = 1,2,3,...)
λ = Wavelength

Rearranging this equation, the wavelengths of standing waves which can be set up are given by

$$\lambda = 2L/n \tag{2.2}$$

Therefore, a standing wave will be set up when the wavelength is equal to twice the length of the air column (λ=2L). Successive standing waves will then form at λ=L, λ=2L/3 etc.

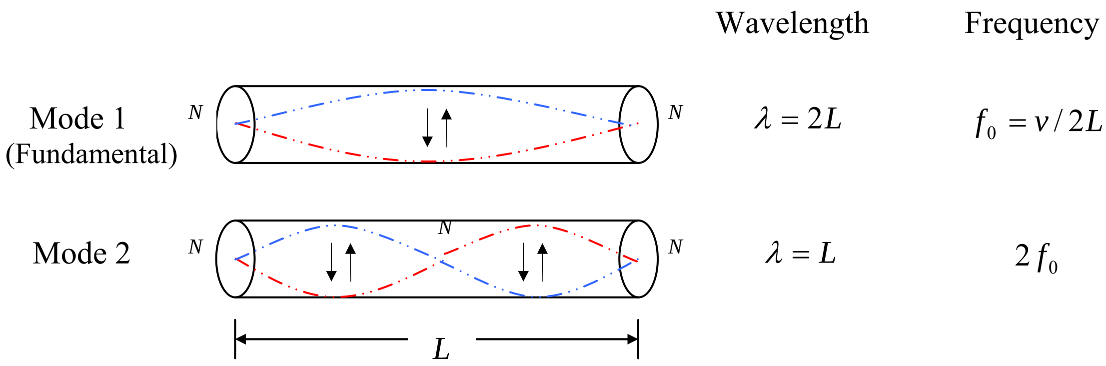
Since $v = f\lambda$, the resonant frequencies of the air column are given by

$$f = nv/2L \qquad (n = 1,2,3,...). \tag{2.3}$$

v = Speed of sound in air

The lowest frequency at which a standing wave will be set up in the tube ($f = v/2L$) is known as the fundamental frequency. At this frequency, a standing wave is generated with pressure nodes at each end of the tube and a single pressure antinode halfway along the tube. Standing waves will also be set up at integer multiples (harmonics) of this fundamental frequency. These standing waves, as well as having pressure nodes at each end of the tube, will also have a number of pressure nodes and antinodes positioned at regular intervals along the tube.

Figure 2-3 shows the first two modal vibrations plotted in terms of pressure. The red and blue curves show the extreme variations in the pressure every half cycle.



L = Length of tube
N = Node
f_0 = Fundamental frequency
v = Velocity of sound in air

Figure 2-3. First two pressure modes of vibration - open ended tube

Standing waves can also be described in terms of the displacement of the air molecules. The ends are displacement antinodes as the air molecules are free to move in and out of the tube. Adjacent antinodes are separated by half a wavelength so the length of the air column L must again be equal to an integer number of half wavelengths. This means that the displacement and pressure is offset by a quarter of a wavelength and so a displacement node corresponds to a pressure antinode and a displacement antinode corresponds to a pressure node. Figure 2-4 shows the first two modal vibrations plotted in terms of displacement.

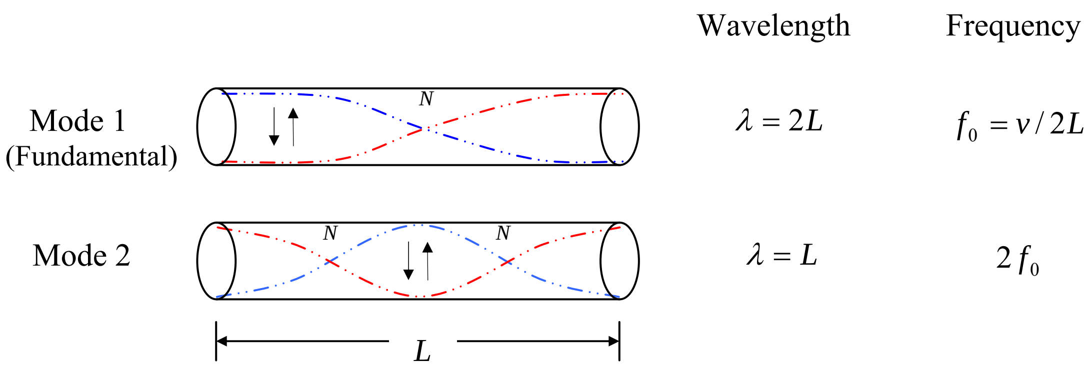


Figure 2-4. First two displacement modes of vibration - open ended tube

2.2.3 Air Column Resonance - Closed End Tube

For a cylindrical tube which is closed at one end and open at the other, the open end will again be a pressure node but the closed end will be a pressure antinode. The distance between a node and a neighbouring antinode is a quarter of a wavelength and therefore the distance between a node and any arbitrary antinode is an odd number of quarter wavelengths. The length L of the air column will be

$$L = n\lambda/4 \tag{2.4}$$

where, in this case, $n = 1, 3, 5,....$

Rearranging this equation, the wavelengths of standing waves which can be set up in an open-closed cylinder are given by

$$\lambda = 4L/n \tag{2.5}$$

Since $f = v/\lambda$ then the resonance frequencies for the tube closed at one end are given by

$$f = nv/4L \tag{2.6}$$

(with $n = 1,3,5,...$).

The lowest frequency (the fundamental frequency) at which a standing wave will be set up in the tube is given by $f = v/4L$. At this frequency, a standing wave is generated with a pressure node at the open end of the tube and a pressure antinode at the closed end of the tube. Standing waves will also be set up odd multiples of this fundamental frequency. These standing waves, as well as having the pressure node at the open end and the pressure antinode at the closed end, will have a number of pressure nodes and antinodes at regular positions along the tube.

Figure 2-5 shows the first two modal vibrations plotted in terms of pressure. The red and blue curves show the extreme variations in the pressure every half cycle.

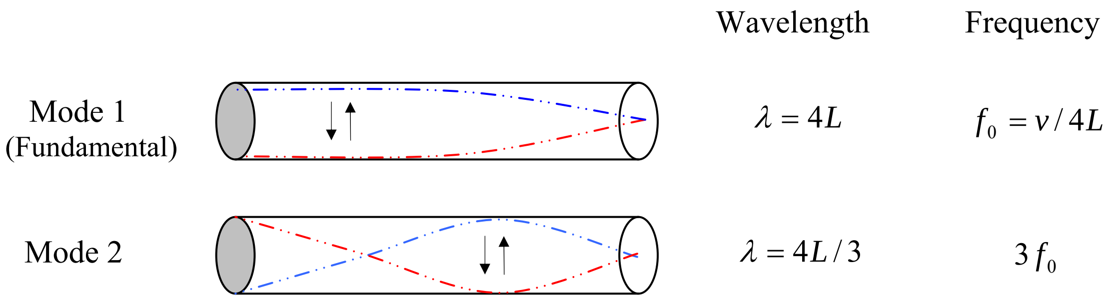


Figure 2-5. First two pressure modes of vibration - closed end tube

Once again the normal modes of vibration can also be plotted in terms of displacement (Figure 2-6). Here the open end will be a displacement antinode as the air molecules are free to move in and out of the tube and the closed end a displacement node.

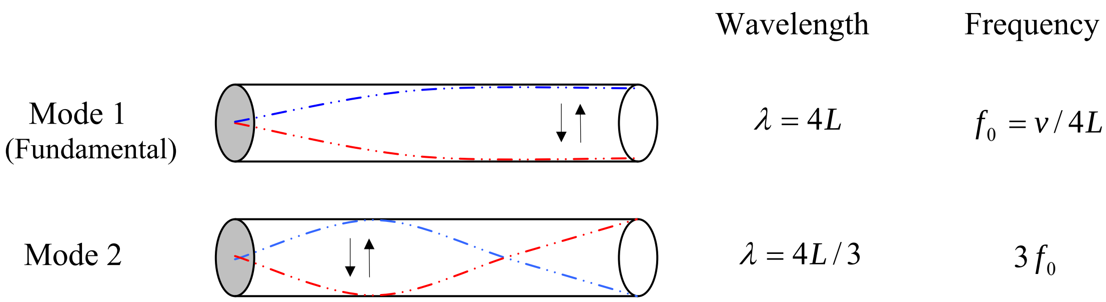


Figure 2-6. First two displacement modes of vibration - closed end tube.

2.3 Brass Instrument Excitation Mechanism

To instigate air column resonance, energy must be imparted to the air column at one end of the pipe. For brass instruments, this initial stimulus comes from a player blowing air through vibrating (or 'buzzing') lips, via a cone/cup shaped mouthpiece, into the air column. The lips act as a control oscillator, introducing an oscillating component of both flow and pressure into the air stream.

When a player goes to play a note, the lips are initially closed. The air pressure is increased in the mouth behind the lips until they are pushed apart. A high pressure pulse of air is released into the instrument. The muscular tension of the lips then causes them to shut as the mouth pressure drops behind them. The air pressure then increases in the mouth again and the process repeats. The rate at which the lips open and close (and hence the rate at which high pressure pulses are injected into the air column) depends on the degree of muscular tension (the player's embouchure); high tension giving a high rate, low tension giving a low rate.

Initially the player adjusts their embouchure so that their lips open and close at a rate close to one of the air column's resonance frequencies. Pressure pulses propagate down the air column until they reach the open end where they are strongly reflected. They return to the mouthpiece where they are again reflected, because the aperture formed between a player's lips when blowing an instrument is much smaller than the instrument's bore. That is, the lips act as a closed end. After a short while, the strong coupling between the lips and the air column ensures that the rate at which the lips open and close matches the resonance frequency almost exactly. Standing waves are set up within the air column and the instrument sounds. To produce higher notes, the player adjusts their embouchure and breath pressure to excite the higher vibrational modes.

2.4 The Effect of Instrument Structure on Air Column Resonances

If a brass instrument consisted simply of a long cylindrical length of pipe closed off at one end by a musician's lips, it would only play on the odd harmonics. However, instruments are musically most useful if the frequencies of the modes of vibration, at least approximately, form a full harmonic series. Consequently, although most brass instruments contain some cylindrical sections of tubing, they also contain sections with more complex geometry. These sections have the effect of shifting the resonance frequencies so that they are close to forming a complete harmonic series.

2.4.1 Function of the bell flare

Modern brass instruments such as the trombone and trumpet require a section of tubing with a constant diameter to incorporate the slide and valve section respectively. The shape of the remaining pipe bore then consists of an expanding flare (the instrument bell). The relative lengths of these sections vary between the different brass instruments.

Adding a bell flare to an instrument has two important effects. Firstly, it is designed to contain acoustical energy within the instrument in order to set up standing waves at specific frequencies. Secondly, adding a bell produces a much louder and clearer tone than produced by a cylindrical pipe only.

In a straight length of pipe all standing wave modes are radiated from the same point, the end of the pipe. The flare, an increase in diameter towards the open end of the tube, has the effect of raising the frequencies of all the modes of vibration, but in particular those of the lower resonances. The reason for this is that, due to the geometry of the bell, waves of longer wavelength are only able to penetrate a short distance into the bell before they are reflected. Shorter wavelengths can travel much further into the bell.

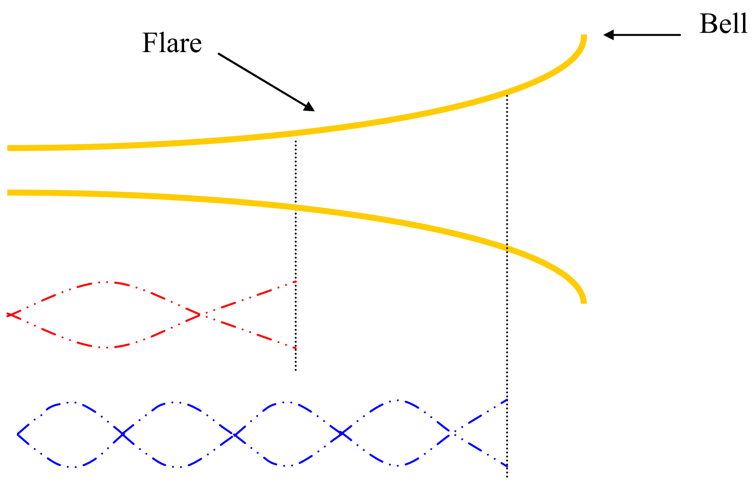


Figure 2-7. Diagram showing relative points of reflection for different wavelengths in a bell flare

The introduction of a bell therefore makes the modes reflect at different points. The effective length of the air column is shorter for the lower modes than it is for the higher modes. As a consequence, the frequencies of the lower modes are raised. This has the effect of bringing the frequencies of all the modes from the second mode upwards into an approximately harmonic relationship.

The fact that waves of shorter wavelength travel further into the bell means that the higher frequencies are more efficiently radiated, with almost complete radiation when the wavelength is comparable with the curvature of the bell. Because these higher frequencies fall in the range where the human hearing is most sensitive, adding a bell to an instrument also has the effect of increasing the loudness of the instrument.

The disadvantage of the bell to the player is that the efficient radiation of the higher frequencies leads to lower reflection and weaker standing waves. This causes higher notes to be more difficult to play than pitches in the middle register (the quality of the resonance or how sharply "notched" in pitch it is depends on the degree of reflection).

The bell also gives the instrument a much more directional characteristic. If the wavelength of sound passing through an opening is much larger than the diameter it will spread out in all directions practically uniformly. This is the case for the low frequency harmonics of the instrument where the wavelengths are much larger than the bell diameter. At higher frequencies the diffraction effects are less pronounced on the smaller wavelengths, the sound concentrated along the instrument axis.

The relative strengths of the spectral components of the sound that escape the instrument bell determine the timbre of a sustained sound and so the bell shape has a primary importance on instrument quality and timbre. Size, how sharply flared, diameter and taper of the bell throat strongly influence tone, intonation and response.

2.4.2 Function of the mouthpiece

The brass mouthpiece comprises a small conical or cup-like section that encloses a volume of air, a narrow constriction (the throat) and a short taper (the back bore) which is of considerably narrower diameter than the instrument bore it widens out into. The mouthpiece acts as a Helmholtz resonator, having one major resonance whose frequency depends on the enclosed volume of air and the geometry of the constriction. A mouthpiece provides a comfortable interface between the player and instrument, giving a musician much more control over the lip buzzing frequency.

Adding a tapered mouthpiece to a closed end acoustic resonator has the effect of lowering the frequencies of the higher modes of vibration. This is because adding a mouthpiece to a pipe will effectively lengthen the pipe bore.

When a brass player plays on the lower resonances (that is, when they excite the lower modes of vibration), where the wavelengths of the sound waves are long compared to the size of the mouthpiece, the amount of lengthening of the pipe caused by the addition of the mouthpiece is equal to the length L_0 of the pipe which would have the same volume V as the mouthpiece as shown in Figure 2-8 (Campbell and Greated, 1987).

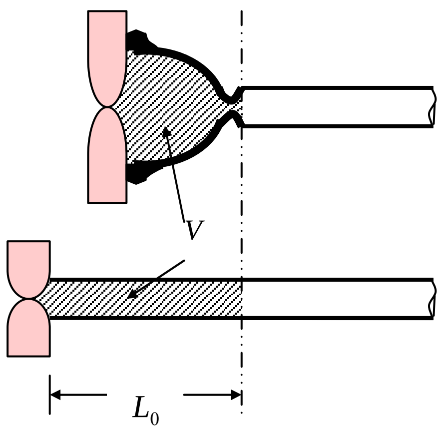


Figure 2-8. At low frequencies addition of a mouthpiece of volume V is equivalent to extending the pipe by a length L_0 giving the same increase in volume (Campbell and Greated, 1987)

As the pitch of notes produced by the brass player is raised (that is, as they begin to excite higher modes of vibration), the effective lengthening of the pipe becomes greater than L_0. When the playing frequency matches the resonance frequency of the mouthpiece f_R, there is a pressure antinode at the lips and a pressure node at the throat. The mouthpiece now behaves as if it were an extension of the pipe of a length $L_R = \lambda/4$ (shown in Figure 2-9), where $\lambda = c / f_R$.

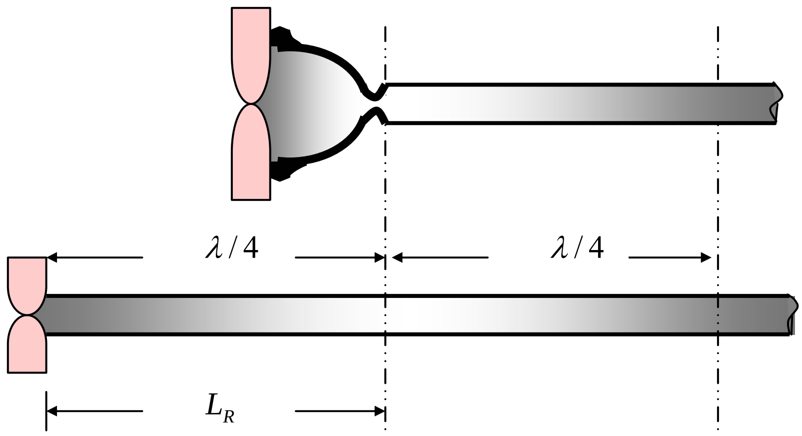


Figure 2-9. At the mouthpiece resonance frequency the pipe is effectively extended by a quarter of a wavelength (Campbell and Greated, 1987)

To sum up, the effective length of the pipe bore increases as the playing frequency increases, reaching a maximum just above the mouthpiece's resonance frequency. The effective length then decreases again. As a result of this effective increase in length, the frequencies of the higher vibrational modes of the air column are shifted down from their original values.

As well as changing the resonance frequencies of a brass instrument, the addition of a mouthpiece also alters the amplitudes of the instrument's resonance peaks, with those peaks which are close in frequency to the mouthpiece resonance frequency being boosted.

As increasing the heights of the peaks makes the corresponding pitches easier to play, adding a mouthpiece has the effect of making notes in the middle register of the instrument more achievable.

The combined effect on the resonance frequencies and amplitudes of the mouthpiece and bell flare is to provide the player with greater feedback from the instrument, enhancing their ability to find a particular note. The frequencies are not truly harmonic nor are notes in the scale of equal temperament but they are close enough for a player to 'lip' up or down the frequency with subtle lip changes.

2.4.3 Altering the Instrument's Effective Length

The frequencies at which standing waves can be set up in a brass instrument (and, hence, the note pitches that can be produced by the instrument) are determined by the speed of sound in air and the length of the pipe. For air columns of a fixed length, standing waves can only be produced at integer multiples of a single fundamental frequency. As a result, only pitches which are (approximately) harmonically related to the fundamental pitch of the instrument can be played. By introducing a means of altering the length of the air column, this fundamental frequency can be changed and, hence, the range of notes available increased.

'Natural' horns and trumpets, which have a fixed length, are played by exciting the different resonances of the instrument. That is by tightening the lips and increasing the blowing pressure to change the mode of the standing wave. Their playing frequencies are close to those of a complete harmonic series and depending on the shape of the bell they can go up to rather high frequencies. Prior to the invention of valves, notes of the harmonic series were the only ones available on the natural instrument.

Because our sense of pitch depends logarithmically on the frequency, successive notes of a harmonic series get progressively closer in pitch. This is exploited by the natural trumpet which is sufficiently long (and certainly longer than the modern equivalent) that, in the playing range of the instrument, the harmonically related resonances are close enough in pitch for a diatonic scale to be achievable. This allows natural trumpet and horn players to produce scales in spite of the lack of valves.

There are, however, a number of problems associated with playing a natural trumpet. It is not easy to select the correct harmonic with lip tension when they are so close together - the harmonics have to be lipped into correct pitch and the instrument still does not give a chromatic scale. To play in another key, extra lengths had to be added by the musicians who then had to pitch a whole new series of harmonics.

In modern brass instruments, valves or slides allow the length of the air column to be altered, increasing the number of notes available on the instrument. A trumpet has a resonance regime that approximates that of a tube open at both ends. This is due to most of its body tapering to different degrees, only a proportion of the instrument consisting of cylindrical tubing within which is situated three valves combining to produce six different notes. The valves open additional sections of tubing for the air column to pass through, lengthening the instrument and making available notes of the harmonic series of a different fundamental. By utilising the various partials of all the available harmonic series, the instrument is made fully chromatic. For the trombone, a long cylindrical metal slide, folded back on itself due to its length for ease of playability, is employed to shorten or lengthen the instrument. Tightening the lips and increasing the blowing pressure produces different harmonics, while moving the slide lengthens the tube lowering the pitch in a continuous range. Combining the different harmonics with the different slide position means a full chromatic scale can be played.

2.5 Concluding remarks

This chapter gives a basic understanding of how a brass instrument functions and, consequently, an insight into the operation of the simple instrument that is used throughout this study. The following chapters will focus upon the vibrations initiated in the walls of the instrument as a consequence of these processes.

CHAPTER 3

Measurement System - Laser Doppler Vibrometry

3.1 Introduction

Laser Doppler Vibrometry (LDV) is a non-contact optical technique for determining the velocity and displacement at a point on a vibrating surface. The technique involves directing a laser beam onto a test object and collecting the back scattered light. Vibration of the object surface introduces a Doppler frequency shift into the reflected beam which is used to measure the component of velocity lying along the beam axis.

LDV has a number of advantages when compared with more conventional measurement devices such as acceleration transducers or displacement sensors. It is a remote non-contact technique so there is zero mass loading of the test object. This is ideal for objects whose frequency response would be greatly modified by a transducer's additional mass. Additionally, a small laser spot size (down to 10 μm) gives high spatial resolution; important when making comparisons between different objects which have a small degree of vibrational variation, and when making measurements on curved surfaces. Finally, LDV enables measurement from most surface finishes over a broad frequency and amplitude range with high output linearity and high accuracy.

The speed of set-up, ease of general use and measurement position relocation, make it ideal for the comparison of wall velocities between identical instruments of varying materials where as many variables as possible are to be removed.

3.2 Interferometer Principle of Operation

Laser Doppler Vibrometry is based on the principle of optical interferometry. Figure 3-1 shows a schematic diagram of a directionally sensitive type of interferometer (described as heterodyne) which combined with a laser makes up a Laser Doppler Vibrometer sensor head.

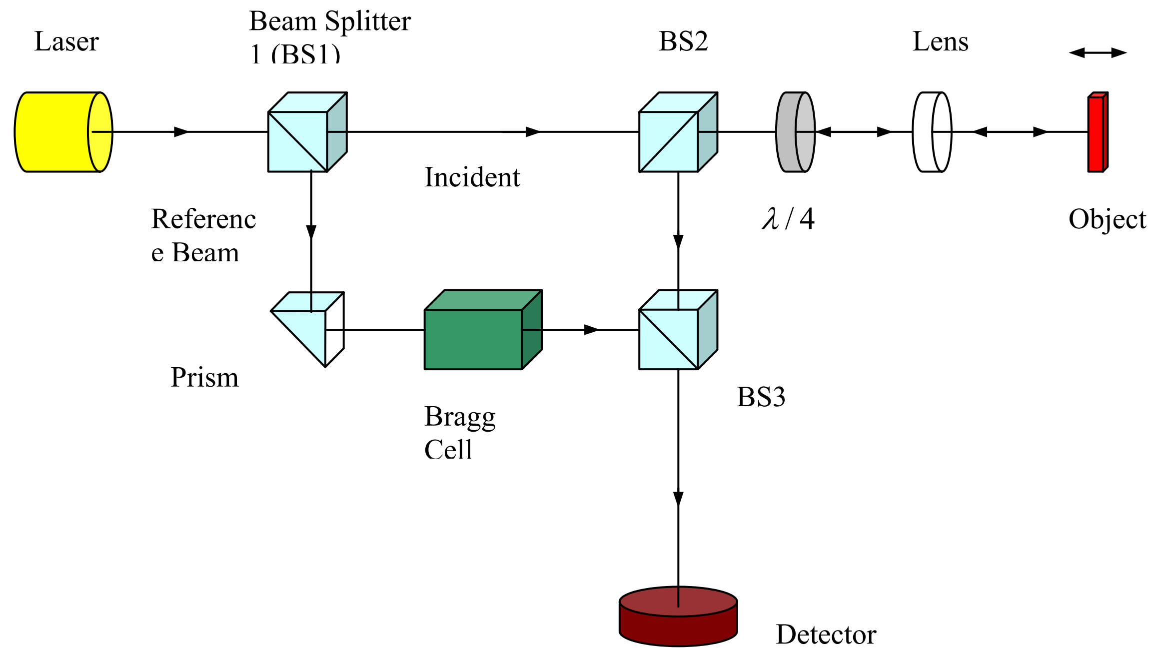


Figure 3-1. Optical configuration of the interferometer (Polytec Scanning Vibrometer PSV-300 hardware manual)

The laser emits a light beam of specific frequency and extremely stable wavelength (0.6328 μm in the case of a helium-neon laser). A polarising beam splitter prism BS1 splits this into an incident beam, to be emitted from the sensor head to the object surface, and a reference beam, which remains within the vibrometer. The linearly polarised incident beam passes through a second polarising beam splitter BS2 and then a $\lambda/4$ waveplate, which converts it to circular polarised light. A lens then focuses the beam on to the surface of the object under test. After reflection, the beam passes back through the lens and is converted back to plane polarised light by the $\lambda/4$ waveplate. The direction of polarisation of the reflected beam is orthogonal to that of the incident beam. The reflected beam is then deflected by beam splitter BS2 to a third beam splitter BS3 where it is recombined with the reference beam causing optical interference. The interference signal is converted into an electrical signal by the photo detectors (two are used to minimise noise and drift) which is then decoded by the electronics controlling the LDV sensor head.

3.2.1 Determination of Velocity

If the incident beam from a vibrometer is focussed on to a moving surface, it experiences a Doppler frequency shift f_D upon reflection from that surface. The frequency of the reflected beam is higher than that of the incident beam if the surface is moving towards the vibrometer and lower if the surface is moving away. This Doppler frequency shift is a function of the velocity component v in the direction of the incident beam:

$$f_D = 2 \cdot \frac{v}{\lambda} \tag{3.1}$$

where λ is the laser wavelength.

If the reflected beam were combined with the incident beam, a beat frequency or intensity modulation equal to the magnitude of the Doppler frequency shift would be generated in the resultant beam. If this resultant beam were measured using photo detectors, the electrical signal produced would fluctuate at a frequency

$$f_{elec} = |f_D| = 2 \cdot \frac{|v|}{\lambda} \tag{3.2}$$

Analysing this signal would only yield the magnitude of the velocity of the moving surface.

For this reason, in a vibrometer head, the reflected beam is combined with a reference beam which has been passed through a Bragg Cell. The Bragg Cell adds a fixed frequency shift f_B to the reference beam (this frequency shift is usually equal to 40 MHz). As a result, the beat frequency generated when the reflected and reference beams are combined is equal to the magnitude of the Bragg frequency shift plus the Doppler frequency shift. So, when the resultant beam is measured using photo detectors, the electrical signal produced fluctuates at a frequency

$$f_{elec} = \left| f_B + f_D \right| = \left| f_B + 2 \cdot \frac{v}{\lambda} \right| \qquad (3.3)$$

As the frequency shift introduced by the Bragg Cell is known, analysis of this electrical signal yields both the magnitude and direction of the velocity of the moving surface.

3.2.2 Determination of Displacement

Displacement information, when calculated by the integration of the velocity signal, can become inaccurate due to integration errors at low frequencies. As a result, more modern systems include a decoder which enables the displacement to be measured independently of the velocity.

When the reflected and reference beams are combined within the vibrometer sensor head, optical interference occurs producing a fringe pattern. The resulting intensity, *I*, measured by the photo detectors varies with the phase difference $\Delta\varphi$ between the two beams according to the equation

$$I(\Delta\varphi) = \frac{I_{\max}}{2} \cdot (1 + \cos\Delta\varphi) \tag{3.4}$$

The phase difference $\Delta\varphi$ between the two beams arises because the two beams travel different distances before they combine at the photo detectors. The phase difference is related to this path difference ΔL by the following expression:

$$\Delta\varphi = 2\pi \cdot \frac{\Delta L}{\lambda} \tag{3.5}$$

When the incident beam from the vibrometer is incident on a moving surface, the path difference becomes a function of time (i.e. it is ΔL (t)). As a result, the phase difference also varies with time causing the interference pattern to move on the photo detector. The displacement of the surface as a function of time can then be determined by analysing the rate at which fringes pass over the photo detector.

3.2.3 Consideration for Optimal Signal

To obtain an optimal signal prior to taking measurements, two main considerations of the experimental set-up had to be taken into account. Firstly, the operating distance between the sensor head and the target object and, secondly, the condition of the test object surface and its orientation with respect to the beam.

To ensure that the detected signal has the best signal-to-noise ratio:

1) The distance from the sensor head to the object surface must be close to an integral number of laser cavity lengths. A poor signal-to-noise ratio maybe due to the distance between the sensor head and object being halfway between the optimal distances. The light source is a multimode laser in which two modes can exist. Interference between them leads to the intensity of the resultant optical signal varying periodically with the stand off distance. Intensity reaches a maximum if the optical path difference is an even numbered multiple of the laser cavity length. A maximum of visibility is present once per laser cavity length (as the optical path difference is equal to twice the stand-off distance).

2) Sufficient scattered or reflected light must reach the photo detector. When a laser illuminates an optically smooth surface (referred to as a specular surface) the reflected beam must be pointed directly into the receiver aperture to enable a measurement. Thus, the surface must be orthogonal to the optical axis of the optics. If, as with the use of scanning vibrometers, there is an angle between the sensor head and any measurement point, the surface may require treatment to allow sufficient laser light to be reflected back towards the optics.

   When a laser illuminates an optically rough surface (referred to as a non-specular surface), the spot has a ‘granular’ appearance. This is known as a speckle pattern and occurs when height variations on the surface are of an order or greater than the laser wavelength, the numerous reflections causing random interference. When performing measurements on a non-specular surface, as that surface moves there is a variation in the intensity of the light that is reflected back to the photo detectors, depending on whether it originates from a bright speckle or a dark speckle. The result of this is noise on the laser vibrometer output. To overcome this, the signal-to-noise ratio of the system can be enhanced by special treatment of the surfaces under measurement by, for example, painting or applying a special reflecting tape.

3.3 LDV Systems Employed over Research Period

Two LDV systems were utilised over the course of the experimentation. A Dantec 55X single point laser vibrometer system and a Polytec (OFV-303/OFV-3001) single point system, which was later upgraded to full scanning capability (section 3.3.2.3). The two systems operate on the same basic heterodyne interferometer principle described above.

Function and operation of the two systems are described in the next section.

3.3.1 Dantec System

The Dantec 55X laser vibrometer system comprises a 55L66 vibrometer module and Uniphase Model 1125P Helium-Neon (He-Ne) 10 mW Class IIIb laser connected to a 55N12 Frequency Shifter and 55N21 Doppler Frequency Tracking Unit.

Figure 3-2. Dantec 55X Laser Doppler Vibrometry System

An IBM compatible PC using a National Instruments E-Series AT-MIO-16E-2 I/O board as the data acquisition (DAQ) card was used to record the signals output from the Dantec system. The board was controlled by a program written in LabView where the time domain output could be viewed along with the resultant frequency spectrum.

The distance over which the vibrometer is capable of measuring depends on the reflectivity of the surface and the bandwidth of the signal processor. The standard lens can be focused to cover distances between 1.2 m and 20 m, but smaller or greater distances are possible with special front optics. Vibrational measurements can be made over a wide range of amplitudes in the frequency range from DC to 740 kHz and it has a dynamic range of 10e-6 m/s to 1.5 m/s.

3.3.1.1 Dantec System Operation

Focussing, by means of the focussing ring on the lens, can be made visually by looking at the focal spot through an attenuation filter. Additionally, an oscilloscope is connected to the system electronics to monitor the 40 MHz square wave of the frequency shifted laser beam, a clear signal being observed when focused.

To begin using the vibrometer system, the middle of the range of frequencies on the Tracker Unit needs to be selected. If the frequency and amplitude of vibration are known then the velocity can be calculated and the Tracker range selected. A level is selected on the frequency shifter that is approximately half the Doppler frequency range setting selected on the Tracker, the ranges on the two units set together. When the Doppler frequency is within the Tracker range a Lock Detector on the Tracking Unit illuminates and the analogue output will be continuous.

3.3.1.2 Dantec Processing Electronics

With the Dantec system, a vibrometer section is always used in connection with a Frequency Shifter, which supplies an electronic and optical frequency shift, and Tracker, designed for broadband demodulation of Doppler signals. The Tracker has a frequency range of 1 kHz to 10 MHz in seven range segments and its output is a 1-10 V analog signal, proportional (in each range) to the frequency of the input signal. If the vibrometer is first set-up to measure a stationary object, the signal output from the Tracker has a frequency equal to the frequency shift. If a vibration is imposed on the surface, the monitor frequency will be modulated with the vibration frequency. The signal is sent from the detector in the vibrometer section to the shifter where it is electronically downshifted, reducing the 40 MHz optical frequency shift to a selected lower frequency. The outgoing beam is optically frequency shifted by means of a Bragg Cell which, in combination with a frequency shifter produces a variable frequency shift allowing detection of motion with alternating directions of velocity. The resulting output from the signal processor is a signal linearly related to the velocity component in the direction of the vibrometer's optical axis.

The Tracker continuously detects the frequency of an incoming signal, supplied directly by the Shifter, the frequency corresponding to the surface velocity under detection. The Doppler signals are amplified and can be further filtered before being output for further signal processing.

Large vibrations may exceed the slew rate (maximum rate of change of frequency) capability of the tracker making it necessary to shift to a higher range where the slew rate is higher. This is done at the expense of sensitivity.

3.3.2 Polytec System

The system used for the majority of experiments comprised a Polytec standard optical interferometric sensor head (OFV-303) and an electronics controller unit (OFV-3001) which can be seen in Figure 3-3. The sensor head consists of a helium-neon (He-Ne) ClassII laser with an optical output of less than 1 mW, a modified Mach-Zehnder interferometer and a lens system. The controller unit contains a velocity decoder (OVD-01) and a displacement decoder (OVD-20).

The OVD-01 is used for applications in the acoustic frequency range up to approximately 20 kHz with five measurement ranges 10 mm/s, 50 mm/s, 250 mm/s, 1.25 m/s and 10 m/s full scale peak. It has a detectable velocity range of 3 μm to 10 m/s. The maximum operating frequency is 50 kHz with a lower frequency limit of 0 Hz (full DC capability).

Figure 3-3. Polytec controller and sensor head

The OVD-20 displacement decoder has eight range settings with a practical lower amplitude limit of 0.01 μm to maximum amplitude of 40 mm. Specifications for the controller modules can be seen in Table 3-1.

The sensitivity of the sensor head makes it possible to measure stand-off distances from less than 0.1 m to several 100 m from a wide range of surface finishes.

The output signals were recorded on the PC based DAQ card where they were analysed using Matlab software, the programme written within the department.

MODULE	MEAS. RANGE mm/s/V	FULL SCALE OUTPUT mm/s	RESOLUTION μ m	MAX. SIGNAL FREQUENCY kHz	MAX. ACCELERATION g	LINEARITY ERROR %
OVD-01	1	20	0.3	20	150	0.5
	5	100	0.3	50	1,600	0.5
	25	500	0.8	50	8,000	0.5
	125	2,500	1	50	40,000	0.5
	1,000	20,000	2	50	320,000	0.5

MODULE	MEAS. RANGE μ m/V	FULL SCALE OUTPUT mm	RESOLUTION μ m	MAX. SIGNAL BANDWIDTH kHz	MAX. VELOCITY m/s	LINEARITY ERROR %
OVD-20	0.5	0.008	0.002	25	0.06	0.1
	2	0.032	0.008	75	0.25	0.1
	8	0.13	0.032	75	1.0	0.1
	20	0.32	0.08	250	2.5	0.1
	80	1.3	0.32	250	10	0.1
	320	5.1	1.3	250	10	0.1
	1280	20.5	5	250	10	0.1
	5120	82	20	250	10	0.1

Table 3-1. Specifications of the Velocity (OVD-01) and Displacement (OVD-20) modules (Polytec Scanning Vibrometer PSV-300 hardware manual)

3.3.2.1 Polytec System Operation

The beam is focussed manually by means of a focussing ring at the lens. The signal level is shown on the back of the sensor head as a 10-part bar display and a 20-part bar display on the controller. A fully lit display indicates the signal-to-noise ratio is maximal. Measurements can still be made even if none of the bar LED’s are showing, although the output signal contains more noise. If the signal level is low or highly fluctuating the sensor head may be at an unsuitable stand off distance.

The velocity and displacement ranges along with filter settings were set via push button and liquid crystal display, the maximum velocity to be measured not to exceed the respective full scale range, the LED VELOCITY OVER light lighting up continuously when the next highest range was to be selected

3.3.2.2 Polytec Controller Unit

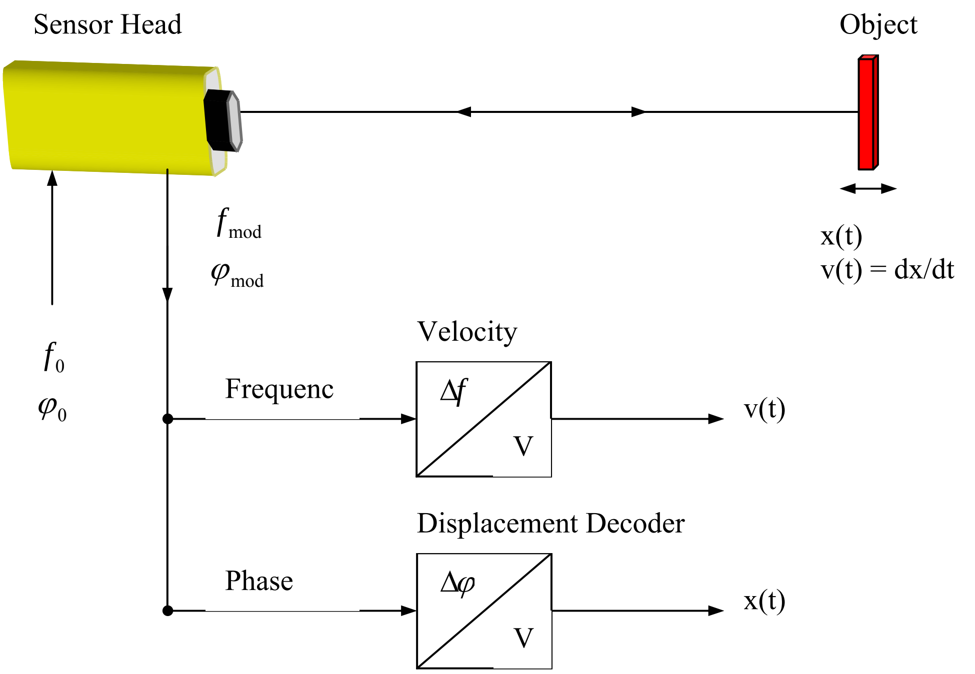


Figure 3-4. Schematic showing signal paths of the vibrometer system (Polytec Scanning Vibrometer PSV-300 hardware manual)

The controller's main function is to demodulate the radio frequency (RF) signal received from the interferometer. The frequency of the signal is the carrier of the velocity information and the phase is the carrier of the displacement information.

The velocity is modulated on the radio frequency of the input signal. In the velocity decoder, an AC voltage is generated which is proportional to the instantaneous velocity of the object with the aid of FM demodulators. In the displacement decoder, optionally the phase of the RF signal is demodulated. An AC voltage is generated which is proportional to the instantaneous position of the object.

The individual demodulators require different reference frequencies, which are in a fixed relationship to the driver frequency of the Bragg cell and synchronized with its driver signal.
The velocity decoder is followed by a low pass filter, which can be set at various cutoff frequencies, that suppresses spurious RF components and limits the bandwidth of the output signal to reduce background noise. The signals then pass through an amplifier, which scales the output signals. The signals are available as an analog voltage. The front end of the controller has additional functions such as signal level display and filters. The settings of the vibrometer are selected via the controller display.

3.3.2.3 Scanning Vibrometer Upgrade

During the period of the research system capabilities were further extended to enable surface scans from a fixed sensor head position. The sensor head was coupled to an OFV-040 Scan Unit housing an optical mirror system allowing a high number of points to be measured consecutively and a VCT-24 live colour video camera with optical zoom for interactive definition of scan points and data overlay. The scan unit contains XY deflection mirrors that automatically steer the laser beam to the desired target on the object with a settling time of <10 ms, a position resolution of 0.0002° and a 40° x 40° field of view.

Scans were controlled by a Polytec purpose written data management system where analogue velocity signals could be displayed as time signals or spectra and the operating deflection shapes animated for each frequency of interest. This gave an accurate visualisation of an object's vibrational characteristics without the inconvenience of further data manipulation and visualisation using separate programs.

3.3.2.4 Signal Conditioning

To optimally drive the various demodulators the RF signal from the sensor head has to be pre-processed. The signal conditioning includes:

- Measurement of the input signal level; providing the user with information on the object's back scattering properties and as an aid to optimally focus the beam.
- Stabilization of the signal amplitude. The input signal level can fluctuate due to different back scattering properties.
- Limitation of the bandwidth. For low velocities only a narrow section of the bandwidth is occupied by the FM (frequency modulated) signal. Only noise is recorded in the remaining bandwidth and is limited by a switchable filter.
- Drop-out reduction via the tracking filter. The back scattered light from the object has a speckled nature (at any instant the detector sees a light or dark speckle). The dark speckles low signal amplitude can lead to loss of signal, or drop-out. This interruption of the signal causes short but high noise signals making decoding the velocity difficult to analyse. Drop-outs are reduced by a tracking filter in the controllers input section. An electronic circuit regenerates high frequency signals based on the principle of the phase locked loop (PLL). The principle is based on replacing the input signal with a distorted amplitude by a stable signal from a voltage controlled oscillator synchronized with the input signals frequency and phase.
- Down-mixing of the frequency. This converts the carrier frequency of the FM signal from 40 MHz originally to lower intermediate frequencies, the process not affecting the modulation content of the signal. The velocity decoder can then work optimally in the

individual measurement ranges. A variable frequency is produced by a local oscillator in a fixed relationship to the Bragg cell's drive frequency.

The oscillator section generates all drive frequencies for operating the other subassemblies in the vibrometer. The drive frequency of the Bragg cell directly determines the optical frequency offset in the interferometer and thus the center frequency's carrier signal. The displacement decoder's phase reference signal φ_{ref} is generated with a fixed relationship to the carrier frequency f_B.

The variable mixing frequency is automatically set by the system control dependent on the selected velocity measurement range.

3.3.2.5 Displacement Decoder

The phase of the signal from the interferometer is the carrier of the displacement information. The information required rides on the phase difference between the driver signal for the Bragg cell and the modulated signal at the photo detector. An object displacement of $\pm\lambda/2$ produces a full demodulation period (a fringe passage) at the photo detector.

The number of fringes counted is a measure of the displacement of the object with accuracy and resolution of $\lambda/2$ which is 316.4 *n*m for the helium-neon laser. The mode of operation in which merely the number of fringes is counted is called the direct counting mode and corresponds to the measurement range of 80 $\mu m/V$. The interferometric phase changes continuously with the object's displacement and displacements of less than $\lambda/2$ can be resolved with electronic interpolation techniques.

System information was obtained from the Dantec 55X Laser Vibrometer Instruction and Service manuals and the Polytec Scanning Vibrometer PSV-300 hardware manual (D-76337).

3.4 Reasons for System Change

Though both systems cover the vibrational ranges required for measurement of the test object the Polytec system was introduced due to concerns over the accuracy of the Dantec vibrometer. For the Dantec, maximum velocity amplitudes measured at a point on the test object's surface were averaged, yet there were still difficulties with focusing and settling at a specific velocity making repeatability of results difficult. With measurements being obtained from one test object, adequate results were obtained to allow mode shape and frequency to be determined, though this was deemed unacceptable if dealing with measurements of small variations between objects. Other direct benefits gained by the Polytec system include increased flexibility of controller functions at the controller display and simpler focusing of the sensor head.

Chapter 4

Background Theory and Preliminary Experiments

4.1 Introduction

In this chapter methods of exciting a cylindrical test object are investigated and the suitability of the LDV system for measuring the response of the test object is appraised. The chosen test object is a length of extruded plastic (polypropylene) pipe. Structural resonance frequencies obtained from experiments are compared with theoretical predictions from standard equations. These demonstrate that the chosen method of mechanical excitation, when used in combination with the LDV system, is highly suitable for the investigation of wall vibrations in simple brass instruments.

4.2 Dynamics of Cylindrical Shells

In this section the basic theory of the dynamics of cylindrical shells is reviewed. Natural and forced vibrational responses, the concept of resonance and the material properties that influence the frequencies at which a structure will resonate are introduced. Next, the possible types of motion, or modes, in which a cylindrical shell structure may vibrate, and the various boundary conditions that determine them are described. Finally, empirical formulae that will provide theoretical values for the resonance frequencies for particular modes are given.

4.2.1 Natural and Forced Response

All structures respond in a particular manner when subjected to impulsive or continuous periodic forces. The frequencies at which a structure will preferentially respond, i.e., with an increased amplitude (be it displacement or velocity), are known as resonance frequencies. (In this work we are concerned with velocity resonances.) These frequencies are determined by the physical properties of the system, i.e. mass, elastic properties, its geometry and the way it interacts with its environment. Together these last two are known as the system's boundary conditions.

When the excitation is impulsive, that is the object is struck sharply, the applied force is broad-band, i.e. contains many (ideally all) frequencies. Damping in the system causes energy at frequencies other than the resonance frequencies to be quickly dissipated, leaving the system to vibrate strongly at particular frequencies. These frequencies are known as the natural resonance frequencies.

Alternatively, if the applied force is continuous and periodic (in the simplest case just a pure sinusoid) the system will only respond strongly when the excitation frequency matches, or is close to a natural resonance frequency. The system will respond less strongly if the driving frequency is some way from a natural frequency. By measuring the resulting reaction of an object to an applied a force, whether it be impulsive or sinusoidal, it is possible to determine its resonance frequencies, because they are the frequencies at which the dynamic response is a local maximum.

For a simple, ideal, one dimensional system, such as a mass, *m*, on a spring of stiffness, *k*, and with damping, *R*, it is possible to set up and solve the equation of motion (Equation 4.1).

$$m\ddot{x} + R\dot{x} + kx = Fe^{i\omega t} \tag{4.1}$$

where *x* is the displacement, *t* is the time and ω is the angular velocity.

In this way, it is possible to predict the frequency at which the system will resonate. The natural resonance of the undamped oscillator (i.e. when *F*=0 and *R*=0) is given by

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{k}{m}}\,. \tag{4.2}$$

Similarly, for a undriven, damped oscillator (i.e. when *F*=0 and *R*≠0) it is given by

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{k}{m} - \frac{R}{2m}} \tag{4.3}$$

When a simple, damped mass on a spring is driven by a sinusoidal force (i.e. when *F*≠0 and *R*≠0), the system's complex displacement response as a function of $\omega\,(=2\pi f\,)$ is given by

$$\tilde{x}(\omega) = \frac{Fe^{i\omega t}}{i\omega[R + i(\omega m - k/\omega)]} \tag{4.4}$$

Similarly, the complex velocity response is the time derivative of Equation (4.4) and is given by

$$\tilde{u}(\omega) = \frac{Fe^{i\omega t}}{[R + i(\omega m - k/\omega)]}. \qquad (4.5)$$

The mechanical impedance of the system is defined as $\tilde{Z} = \frac{\tilde{F}}{\tilde{u}}$ (where $\tilde{F} = Fe^{i\omega t}$) and can be expressed in terms of the resonance frequency, ω_0, mass, *m*, and a quality factor $Q = \frac{k\sqrt{m}}{R}$ as

$$\tilde{Z}(\omega) = (1/i\omega)\left(\omega^2 - \omega_0^2 + i\,\omega_0\omega/Q\right) \qquad (4.6)$$

The form of the displacement, velocity and impedance resonances are clearly determined by the parameters ω_0, *m*, *k*, and *R* (or *Q*), which quantify the resonance frequency, mass, stiffness, damping of the system. Consequently, these are sometimes referred to as the *modal parameters*. Examination of Equation (4.6) shows that *m* (along with terms pertaining to the damping) is primarily responsible for the amplitude of the response, *Q* (along with terms pertaining to the damping) determines the width of the resonance, whilst ω_0 clearly defines its location in frequency-space.

However, for a more complicated system such as a cylindrical shell, there is generally no simple equation of motion. Consequently there is no simple analytical expression from which the modal parameters, in particular the resonance frequency, can be obtained.

This difficulty can be resolved by treating the object as a composite mesh of simpler systems, such as masses on springs, whose equations of motion can be solved. The response of each individual system is then allowed to influence its neighbours and in this way the response of the whole object can be determined. This is the underlying method that is employed in Finite Element Analysis (FEA).

4.2.2 Boundary Conditions

The solutions to a given equation of motion (e.g. Equation 4.1) are reduced in number by applying boundary conditions appropriate to the object in question. The boundary conditions determine the type of motion that is permitted at the extremities of the object and hence determine its resonance frequencies.

Mathematically, the boundary conditions amount to practical restrictions on the nature of the motion of the object. For instance neighbouring portions of the object cannot have displacements of $+\infty$ and $-\infty$. Similar restrictions may also apply to the gradient of the displacement, most commonly that the gradient is smooth and continuous, i.e. doesn't suddenly flip from +ve to -ve.

Similarly, there are common restrictions on the allowed combinations of the displacement and gradient. From the point of view of beams, plates, membranes and shells these are summarised in following three cases, which are known as the "free", "hinged" and "clamped" conditions:

(i)	free end/edge	x≠0	$\frac{dy}{dx} \neq 0$
(ii)	hinged end/edge	x=0	$\frac{dy}{dx} \neq 0$
(iii)	clamped end/edge	x=0	$\frac{dy}{dx} = 0$.

Applying certain boundary conditions to a particular system determines which set of solutions to the equation of motion are possible and hence which set of frequencies the system can resonate at.

4.2.3 Material Characteristics

As mentioned in the previous section the material characteristics, density (mass), stiffness and damping, have a profound effect on the mechanical response of a system. The material characteristics determine the speed of sound in the material and the degree of damping a wave or vibration will experience as it propagates through an object. Perhaps the most obvious material property is its density. However there are other material characteristics that describe how a material responds in particular directions with respect to the applied forces. These are known as the elastic modulus and Poisson's ratio.

Strain is the fractional change in length divided by the original length

$$strain = \varepsilon = -\frac{\Delta l}{l}. \qquad (4.7)$$

Stress is the applied force unit area

$$stress = \sigma = \frac{F}{A}. \tag{4.8}$$

The strain and the stress along the *x* direction are related by the Young's Modulus, E_x, in that direction via Hooke's law

$$\frac{F_x}{A} = -E_x \frac{\Delta l_x}{l_x}, \tag{4.9}$$

so long as the strain and stress are small. This can be re-expressed in the more familiar form as

$$F_x = -k_x \Delta l_x \tag{4.10}$$

where the stiffness $k_x = E_x A / l_x$. The minus signs in both Equations 4.9 and 4.10 ensure that the Young's modulus is a positive quantity for most materials.

Poisson's ratio completes the description of the response of a material to applied forces. It is the quotient of two strains along different directions and can therefore reflect the fact that, for most materials, the application of a force in one direction results in a change in size of the sample in another direction (e.g. many materials shrink in the plane perpendicular to which they are stretched). Poisson's ratio for the axes *x* and *y* is defined as

$$\upsilon_{xy} = -\frac{\Delta l_x / l_x}{\Delta l_y / l_y} = \frac{\varepsilon_x}{\varepsilon_y}, \tag{4.11}$$

where $x \neq y$.

It is clear that most materials and structures do not have the same mechanical properties in all directions, i.e. they are not necessarily homogeneous and isotropic. For this reason the elastic properties of materials, i.e. the Young's Moduli, are often cast in tensor form and are given as E_{xyz}, where the x, y and z signify the three principal orthogonal axes. These material tensors comprise nine terms, however for the majority of known materials they are diagonal in nature, meaning that $E_{xy} = E_{yx}$ thereby reducing the number of numerically different terms significantly. However, the materials and geometries that will be considered in this work permit further simplification of the material properties to the simple, homogeneous, isotropic case without any difficulty. In other words $E_{xyz} \rightarrow E$ and only one value is required.

Finally, by differentiating and rearranging Equation (4.9) the equation of motion can be cast into a form which is comparable with a standard wave equation and it becomes apparent that the phase speed, or speed of sound can be expressed in terms of the Young's Modulus and mass per unit element, or density as

$$c = \sqrt{\frac{E}{\rho}}. \tag{4.12}$$

We therefore see that the material properties, i.e. Young's Modulus and density, directly determine the speed of sound in the object and hence determine the resonance frequencies that can be supported by a given structure made of a given material.

4.2.4 Mode Frequencies

The existence of damping in a dynamic system is almost inescapable and results in a smearing of the solutions for ideal, non-damped systems such that the resonances are broadened and spread over a range of frequencies. For complex systems, whose equations of motion may have solutions that are closely spaced in frequency, this may mean that stronger resonant peaks may dwarf and obliterate weaker resonances. It may therefore not be simply that all resonances result in locally heightened velocity (or displacement) responses. Therefore, from an experimental point of view, this may mean that identifying the peaks in a measured response may not be sufficient. However, systems that have straightforward geometries tend to be relatively well-behaved in this respect, i.e. separated in frequency-space. In such cases the mode frequencies can be treated as those frequencies which yield a local peak response.

Given the material properties and the geometry of the system one can attempt to solve analytically the equations of motion in cylindrical polar coordinates (or any appropriate coordinate system) for a given set of boundary conditions to find the resonance frequencies. This, however, turns out to be far from trivial to do. Consequently the formulae below, that give the resonance frequencies f_{ij} (the meaning of the subscripts i, j will be described in more detail in the following section) of a cylindrical shell clamped at each end, are somewhat empirical and must be used with due care. The following expression is really a more advanced form of $f = \lambda/c$ where c is now expressed in terms of the material parameters E, ρ and ν as well as an effective pipe radius R. Hence

$$f_{ij} = \frac{\lambda_{ij}}{2\pi R}\sqrt{\frac{E}{\rho\left(1-\nu^2\right)}}. \qquad (4.13)$$

In this expression R is the mid-surface radius of the cylindrical shell (the average of the inner and outer radii), ρ is the shell wall's density, E is the Young's Modulus of the wall material, ν is Poisson's ratio of the wall material and λ_{ij} is a dimensionless parameter related to wavelength (see Section 4.2.5). The indices i and j respectively index the number of circumferential waves and axial half-waves of the mode shape in question. In the next section the various different types of mode shape will be identified along with the indices used to describe them (Blevins, 1979).

4.2.5 Mode Shapes

This section deals with how the response of the structure varies over its extent. When only one frequency component is present and this frequency is a resonance frequency, the resulting spatial response of the system is known as a mode shape. When more than one frequency component is present (and these need not necessarily be resonance frequencies) the resulting spatial response is known as an operating deflection shape or ODS. As the response at frequencies other than resonance frequencies tends to be significantly less than at resonance frequencies, ODSs tend to made up of superpositions of mode shape components (not necessarily all in the same proportion, i.e. some modes may be more strongly excited than others).

As discussed in Section 4.2.3, the modes shapes of a system are determined by the boundary conditions and geometry of the system. Similarly, the nomenclature used to describe and index the modes also relates to the intrinsic geometry of the system, be it rectangular, cylindrical, spherical or akin to some other orthogonal curvilinear coordinate system. The system that we will be dealing with throughout this work is a duct or a cylindrical shell. Consequently, the mode shapes are described according to a cylindrical geometry.

The principal mode types are bending modes along the major axis of the duct, radial modes, axial modes, torsional and modes that are a combination of radial, axial and or torsional modes of vibration. Examples of these are illustrated in Figure 4-1.

Values of the dimensionless parameter, λ_{ij}, for the different mode shapes are given by the expressions below.
All other variables are as defined as following:

$i =$ Number of circumferential waves in mode shape
$j =$ Number of axial-half waves in mode shape
$\mu =$ Density of cylinder materials
$E =$ Modulus of elasticity
$R =$ Cylinder radius to midsurface
$h =$ Cylinder thickness
$L =$ Effective length of cylinder
$\nu =$ Poisson's ratio

(1) Bending Modes (Blevins, 1979)
These are modes in which the duct flexes along its length.

$$\lambda_{ij} = \frac{j^2 \pi^2 R^2}{L^2} \sqrt{\frac{(1-\nu^2)}{2}} \tag{4.14}$$

i=1, j=1, 2, 3… (Blevins, 1979)

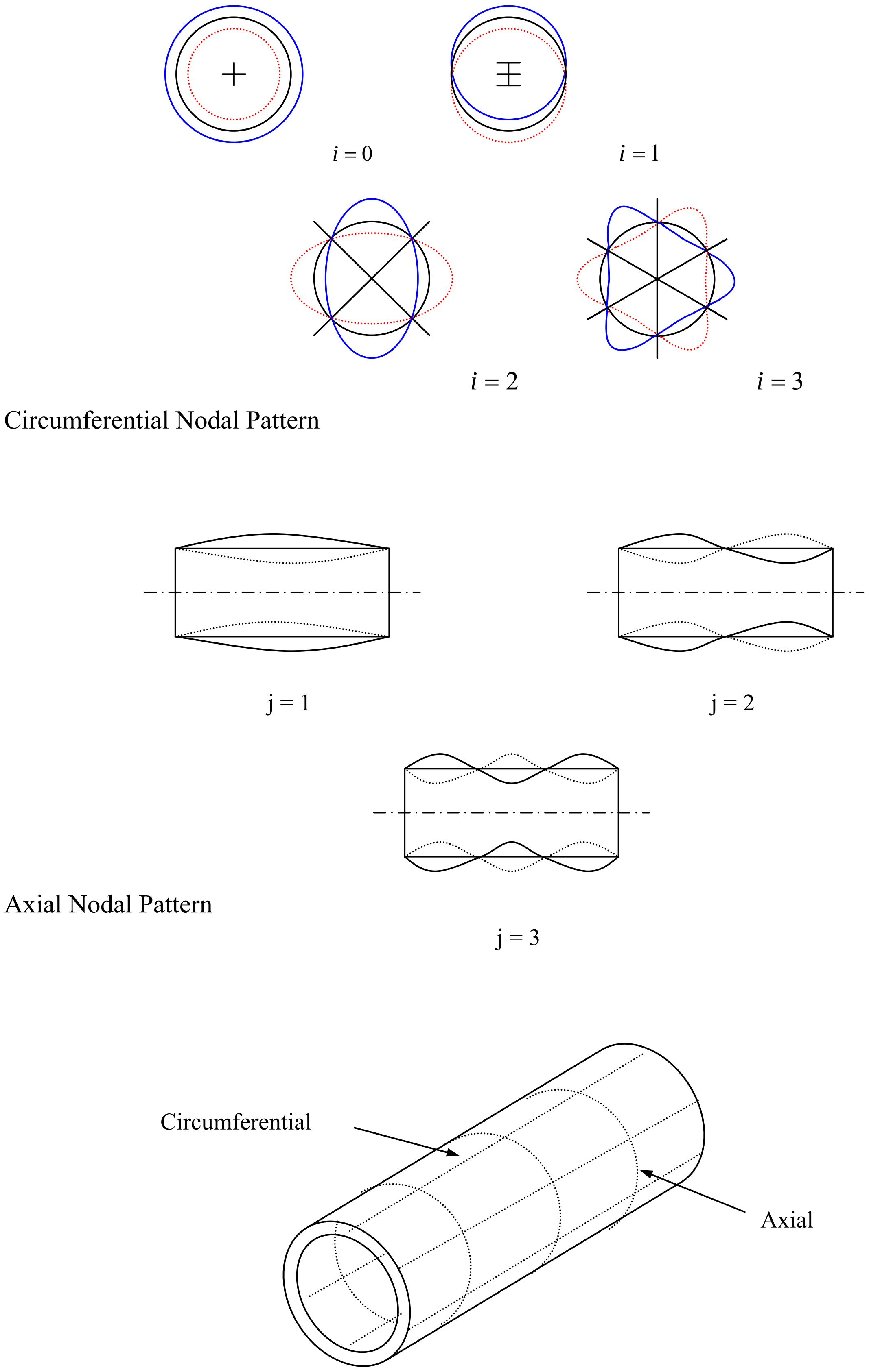


Figure 4-1. Mode shape figures (Blevins, 1979).

(2) Radial Modes

These are modes in which the duct's radius varies. They are sometimes referred to as "breathing modes" and result in a significant change in volume of the duct on account of the variation of the radius through each cycle.

$$\lambda_{ij} = 1$$

i=0, j=1, 2, 3… (Blevins, 1979)

(3) Radial-Axial Modes (Blevins, 1979)

These are modes that combine both radial and axial variation.

$$\lambda_{ij} = \frac{\sqrt{\left\{\left(1-\nu^2\right)\left(\frac{j\pi R}{L}\right)^4 + \left(\frac{h^2}{12R^2}\right)\left[i^2 + \left(\frac{j\pi R}{L}\right)^2\right]^4\right\}}}{\left(\frac{j\pi R}{L}\right)^2 + i^2} \tag{4.16}$$

i=2, 3, 4… j=1, 2, 3… (Blevins, 1979)

(4) Torsional Modes

These are modes in which the duct undergoes a twisting motion around its principal axis. They were not studied in this project.

The bending modes of a cylindrical shell (pipe) only will be studied in this work as other mode types are not activated until much higher frequencies. The first two bending modes are portrayed in Figure 4-2.

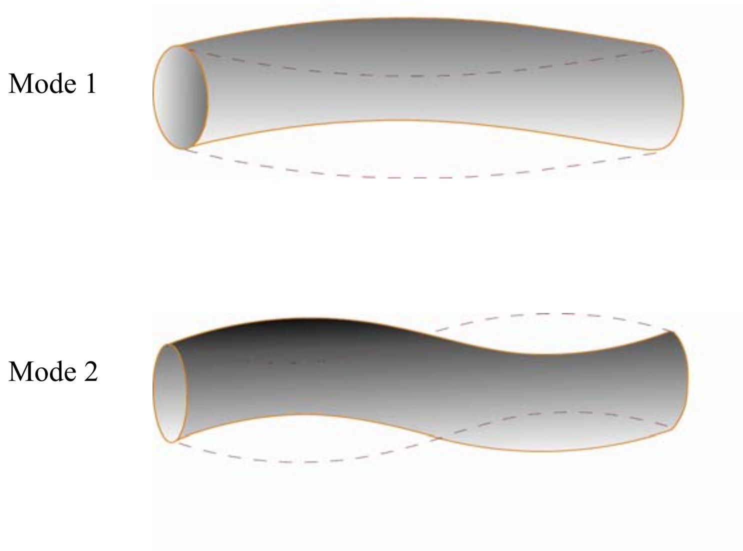


Figure 4-2. First two bending modes of a cylindrical shell.

Mode shapes of structure can be measured by exciting the structure at a fixed location and using a roving transducer to measure the response (known as a transfer response measurement). Alternatively, they can be determined by measuring the normalised response at the point of excitation and roving this point over the structure (an input response). Either the input force is impulsive, i.e. broadband, or the input force is monochromatic and swept over a frequency range. In both cases these measurements yield a mechanical response that is a function of frequency from which the modes can be identified by cross comparing the Frequency Response Function (FRF) measured at a range of locations on the structure.

All the terms of the FRF matrix don't have to be measured due to reciprocity. The mass, damping and stiffness matrices that describe the system are symmetric so the FRF matrix is also symmetric - an input excitation at point one and measurement at point four would produce the same result as an input excitation at point four and measurement at point one.

4.2.6 Summary

Section 4.2 presented and briefly discussed the fundamental theory that under-pins the experimental techniques that will be applied in the following section and subsequent chapters. The concept of resonance, of both free and driven structures, resonance frequency, and the modal parameters, m, k, R, ω_0 and Q have been discussed. The response of materials to applied forces, i.e. stresses and strains, and this is quantified by the Young's modulus and Poisson's ratio, has also been introduced. It has also been shown that the density, Young's modulus and Poisson's ratio determine the wave speed and hence the resonance frequencies that are possible on a given structure. The importance of boundary conditions and how the geometry of the system determines the mode shapes that can be supported on a given system has also been discussed. Finally, focusing on a cylindrical shell, the principal mode shapes of interest in this thesis have been identified and a nomenclature for describing them as been introduced.

4.3 Excitation and Measurement of Mode Shapes and Mode Frequencies

This section examines some of the problems associated with exciting structures and measuring their mechanical responses to applied forces.

4.3.1 Decoupling the Excitation and Pick-up Transducers

When an excitation or measurement transducer is attached to a structure it becomes part of that structure. This has several important ramifications for the scientist who wishes to determine the modal properties of the structure. Any mechanical transducer has, by its very nature, a resonant response of its own. When the device is attached to the structure the two responses are combined and altered. And so, it becomes important for the scientist to decide whether, when adding the transducers to the structure they significantly modify its mechanical response.

The main ways by which these problems are avoided or reduced is to ensure that the resonance(s) of the transducer(s) is sufficiently removed in frequency from those of the structure and that the effective mass of the transducer is insignificant in comparison with the modal mass(es) of the structure. In this way the mixing of the responses is reduced. Similarly the size and contact area of the transducer can prove to be crucial. These considerations necessitate careful design and choice of transducers on the part of the scientist. The next section briefly surveys a few of the more popular approaches that are employed when measuring modal properties.

4.3.2 Mechanical Exciters

The problems described above have led to the development of a number of novel practical solutions. One way in which these problems have been addressed is by the use of a mechanical shaker with needle like attachment, by which the transducer components are attached to the test object. The needle solves many problems. Firstly, it reduces to a minimum the contact area allowing a response to be measured at a particular point rather than being the average over a larger area. Similarly this reduces the loading of the structure to a point. Secondly, it is relatively easy for the steel needle to be designed so that its resonances are removed in frequency from those of the object, thus reducing contamination. Thirdly, the needle can be made to have a low mass, also reducing contamination.

A stinger attachment is usually included with a mechanical shaker source as this uncouples the effects of the shaker from the object by comparing the input to output force at each frequency input.

Other types of mechanical exciters may be more appropriate in other circumstances, for instance, when structure being tested is large and massive and large excitation forces are required, it may be possible and more convenient to attach a more conventional electro-mechanical transducer to the object.

When the structure is lightweight and relatively small, with low modal masses it becomes more difficult to excite the object and measure the response without risk of contamination. When miniaturisation of stingers and electro-mechanical devices proves problematic nearly-non-contact electromagnetic pickups may prove appropriate.

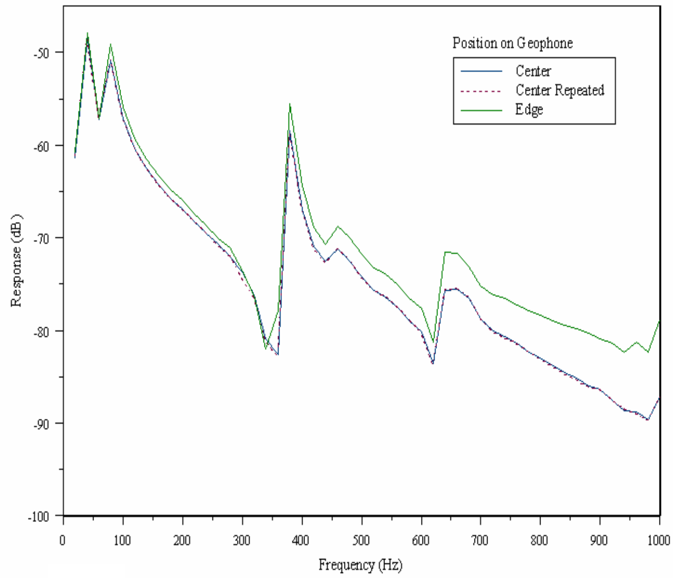
4.3.3 Assessment of the Response of Two Mechanical Exciters

Two mechanical excitation shaker mechanisms were employed in order to ascertain the most effective and repeatable method of exciting a cylindrical pipe wall with a sinusoidal waveform. The first mechanical shaker was a Mark Products Ultraphone Mark II transducer (Figure 4-3(a)). The second mechanical shaker was a Ling Dynamic Systems mechanical shaker Type 261 with a fine pointed needle attachment. This attachment permitted direct excitation at a single point (Figure 4-3(b)).

Figure 4-3. The (a) transducer and (b) shaker excitation mechanisms used in the experimental set up

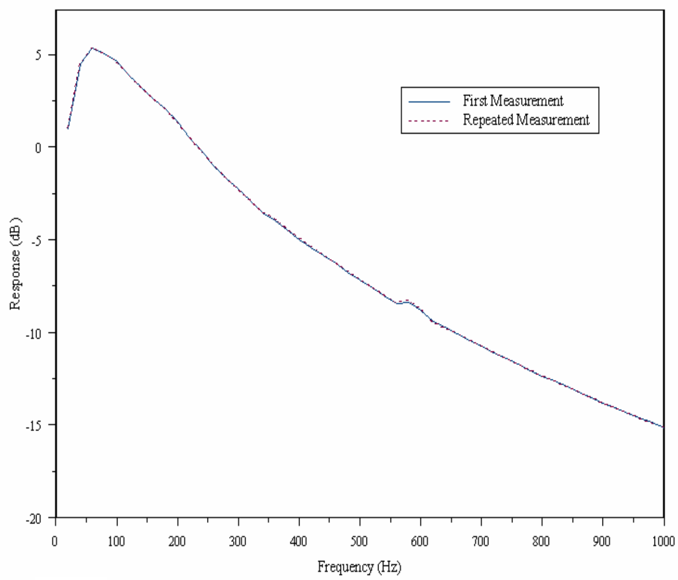
To assess both the response of the transducer and the shaker, the LDV was used to obtain frequency response measurements directly from them. An adequate signal lock was achieved on each by adhering Scotch M3 reflective tape to specific locations on their surfaces (the mechanical shaker had its needle attachment removed). Measurements were taken at two positions on the surface of the transducer's base, at its centre and also midway along its radius.

Mark Products Ultraphone



(a)

Ling Dynamics Shaker



(b

Figure 4-4. Response curves over the frequency range for (a) Mark Products Ultraphone and (b) Ling Dynamics Shaker

Initial tests with the two excitation transducers (the Mark Products Ultraphone and the Ling Dynamics Shaker) in conjunction with the LDV showed that the Ultraphone gave less consistent and less reproducible results than the Ling Shaker, see Figure 4-4.

The response curve for the Mark Ultraphone (Figure 4-4(a)) shows that its response to a change in frequency varies between different locations on the transducer surface. This problem is due to the fact that the device has a relatively large base (diameter of 2.5 cm), and hence large contact area with the pipe. Consequently, it does not give a point measurement. Indeed Figure 4-4(a) shows it was giving different responses at different contact points. The size and shape of the transducer meant it would be awkward to consistently re-position on a pipe and that it would be necessary to use adhesive. The difficulty of attaching, detaching and re-attaching the ultraphone would inevitably introduce errors into the measurements.

The mechanical shaker was chosen as the excitation source for use in subsequent experiments, as its needle attachment would make it easier to apply the excitation force in the same location on successive pipes, thereby giving a higher overall degree of reproducibility and consistency.

In the following section a preliminary experiment is made in which we assess the suitability of the combination of an electro-mechanical shaker with a needle attachment and an LDV system to measure modes of vibration of a test object comprising a length of plastic pipe.

4.4 Application of the LDV Measurement System to Determine the Mode Shapes and Frequencies of a Cylindrical Test Object

In order to assess the effectiveness of the LDV system to measure the response of a cylindrical system, such as a simple musical instrument, an experimental set up was designed to test a section of plastic (polypropylene) piping.

4.4.1 *Experimental Apparatus and Plastic Test Pipe*

The test object used in this series of experiments was a section of cylindrical plastic piping of effective length 47 cm, radius 2.2 cm and with relatively thin walls (in relation to its length) of thickness 0.21 cm. Preparation of the pipe consisted of a narrow strip of Scotch M3 reflective tape being adhered along its axis to enhance the signal-to-noise ratio of the system and to ensure adequate signal lock by the vibrometer.

Measurements on the pipe took place in an anechoic chamber. The vibrometer apparatus was set-up on two anti-vibration optic tables (Ealing Electro-Optics Micro-g Series 62 Post Isolator Systems). Each table consisted of a heavy-duty base of four steel vertical posts onto which an aluminium optical breadboard was placed.

Vibration isolation was provided by pneumatic isolator units attached to the base supplied with compressed air at a constant pressure. It was important that the vibrometer remained stationary. Unwanted movement of the sensor head, due to background noise or building vibration, could result in spurious variations in the optical path length between the LDV and test object which could subsequently result in an incorrect measurement of the surface velocity.

Figure 4-5. Experimental set up for plastic test pipe

Figure 4-5 shows the experimental set up for the plastic pipe experiment. The Dantec vibrometer was positioned horizontally on one anti-vibration table isolated from that of the object under investigation. The plastic pipe was fixed horizontally between two retort stands (which were screwed into one of the anti-vibration optical tables) and rigidly clamped at each end around its entire circumference by jubilee clips preventing movement perpendicular to the length of the pipe (Figure 4-6). The inside of each clip was lined with a layer of foam padding to ensure clamping contact around the entire pipe circumference.

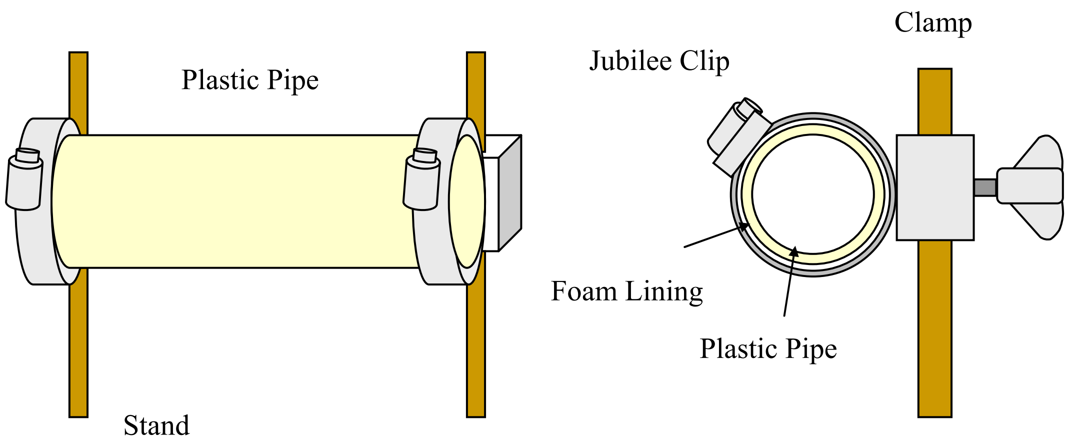


Figure 4-6. Clamping condition at the test pipe ends (plan and end view).

For simplicity and because it allowed comparison with theoretical predictions, both ends of the pipe were clamped in the same manner. So as not to deform the ends of the pipe by over-tightening, two aluminium discs of a width matching that of the jubilee clips were fashioned to fit exactly within the ends of the pipe. Such deformation could easily change the dynamic response of the pipe.

The mechanical shaker used in this experiment was a Ling Dynamic Systems mechanical shaker Type 261 with a fine pointed needle attachment. This attachment permitted direct excitation at a single point. The shaker, rigidly supported by a metal frame, was positioned vertically and close to one end of the pipe and driven using a Thurlby Thandar Instruments (TT*i*) TGA 1230 Sythesised Arbitrary Waveform Generator and Brüel & Kjær Power Amplifier Type 2706.

Due to reciprocity, the mode shape can be extracted by keeping the excitation source at one position and measuring the response at all the other points.

A front surface mirror was fitted in a way as to overhang the plastic pipe but remain isolated from the excitation mechanism. The laser beam was reflected by the front surface mirror, set at 45°, and focussed on to the top of the object pipe in the same plane as the excitation source (Figure 4-5).

4.4.2 Experimental Method

The excitation source was activated and a point by point series of measurements made using the LDV. For each location on the pipe the source was driven at discrete frequencies over a range of 20 Hz to 1.5 kHz, in 20 Hz steps. Measurements were taken at a total of ten locations at 4 cm intervals along the upper edge of the pipe. At each location the resulting output voltage from the LDV was amplified and recorded using a National Instruments data acquisition DAQ card and anti-aliasing filter. A schematic diagram of the experimental system can be seen in Figure 4-7. In order to assess the symmetry of the pipe's response an additional set of measurements were taken along the side of the pipe with the mirror removed and the laser beam directed on to the pipe at ninety degrees to the excitation source.

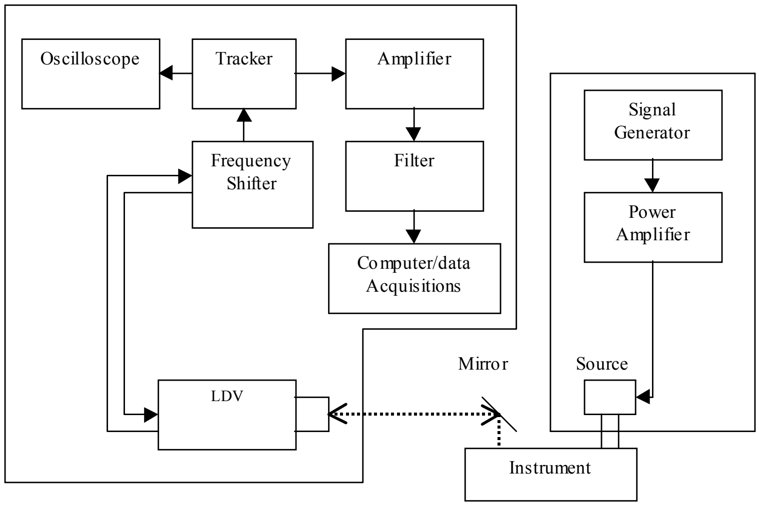


Figure 4-7. Schematic diagram of the experimental system

4.4.3 Results and Discussion

4.4.3.1 Determining the Material Properties of the Plastic Pipe

The Young’s modulus of the material that the plastic pipe was made from was found by performing a tensile test. This was done using an MTS 810 load frame with 858 microconsole servo-hydraulic system. Obtaining a good, accurate result proved problematic as the sample repeatedly slipped from the jaws of the testing equipment. The number of attempts to measure the Young's modulus was also limited by a restricted number of samples of the material.

The density of the plastic was obtained by accurately weighing a small section of the pipe on a mass balance. The volume of the sample was calculated from measurements and the density subsequently determined.

A value for Poisson's ratio was estimated at 0.25, as Polypropylene comprises polymers which act in a non-linear manner when put under tension, thereby invalidating standard mechanical measurement techniques.

	Young's Modulus GPa	Poisson's Ratio -	Density kg m^{-3}
Plastic Pipe	3.74	0.25	1592

Table 4-1. Material Properties of the plastic pipe.

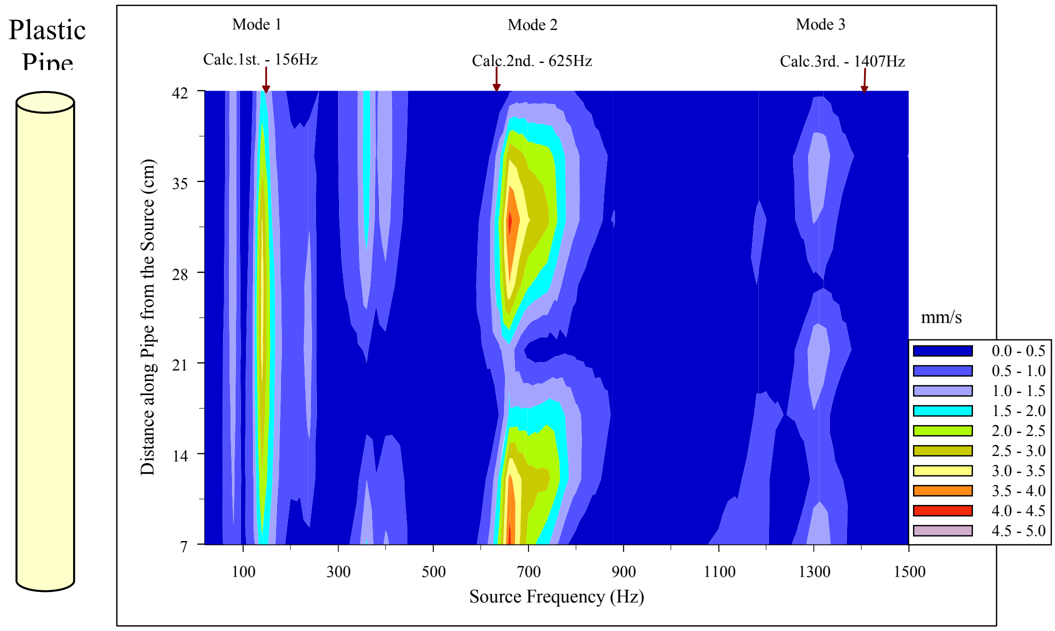
4.4.3.2 Comparing Theoretical and Experimentally Determined Resonance Frequencies

Resonance frequencies for the first three bending modes of the plastic pipe obtained from experiments are presented in Table 4-2 along with theoretical values calculated using equations 4.13 and 4.14.

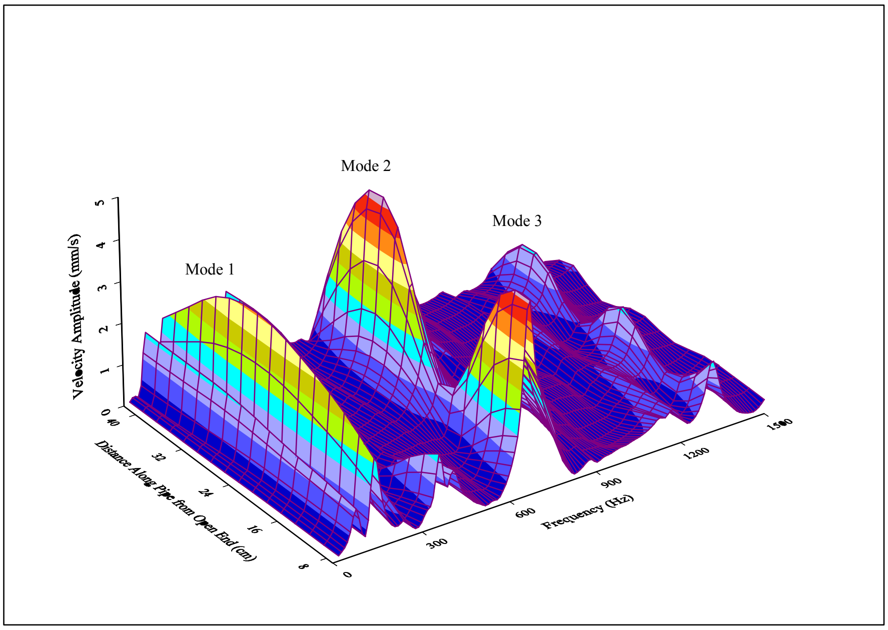
	Mode 1	Mode 2	Mode 3
Experimental Frequency (Hz)	140	660	1300
Theoretical Frequency (Hz)	156	625	1407

Table 4-2. Results of theoretically and experimentally derived mode frequencies

Figure 4-8(a) is 2-D contour plot of the response plotted as a function of excitation frequency and position along its length. The pipe is orientated parallel to the y-axis. This graph clearly shows the allows the variation in peak velocity amplitude at particular



(a)



(b)

Figure 4-8. (a) 2D contour plot and (b) 3D plot of velocity amplitude variation with frequency along the plastic pipe frequencies and positions. The resonant frequencies are clearly distinguishable in the plot, as are the shapes of the first three bending modes.

Figure 4-8(b) is a 3-D representation of the variation in velocity amplitude with frequency and position and clearly depicts the amplitude and shape of the bending modes.

The theoretical frequencies as given in Table 4-2 that correspond to the bending modes identified by the Shaker-LDV experiment are marked on the upper x-axis on Figure 4-8(a) to facilitate comparison. The theoretical values are found to be in reasonable agreement with the experimental measurements, deviating from the experimental frequencies over a range of 5% (mode 2) to 10% (mode 1). This variation is most likely due to a number of experimental issues. Firstly, the pipe wall material may exhibit some degree of inhomogeneity and variability along its length. Secondly, the geometry of the pipe may vary along its length. It may not have a consistent radius or wall thickness. Thirdly, the clamping conditions may not be close enough to the ideal case pre-supposed by the theory. Finally, the frequency resolution (step size between measurements) was quite low (20 Hz steps) and so it is quite likely that the pipe was driven some way from its resonances.

4.5 Conclusion

This chapter has reviewed the fundamental background theory that under-pins the experimental techniques of modal analysis. Section 4.2 introduced the concept of forced and natural response, resonance, resonance frequency, the modal parameters (ω_0, m, k, R and Q), boundary conditions, material properties and their effect on the resonant response of structure and finally we looked at the response of more complex entities such as cylindrical shells, introducing their mode shapes and semi-empirical formulae for their resonance frequencies. Section 4.3 briefly discussed issues relating to excitation and measurement of the dynamic responses of structures and presents some results from experiments evaluating two different excitation transducers. Section 4.4 presented some results from a pilot experiment using the preferred excitation source and the LDV system and results from experiments to determine the modal properties of a length of plastic pipe. Reasonable matches between theoretical and experimental values of the resonance frequencies for the first three structural modes of the pipe were obtained. In conclusion these experiments demonstrate that an LDV system, in conjunction with the Ling Dynamics Shaker and needle attachment provide a satisfactory experimental set-up with which to investigate the wall vibrations of simple brass instruments.

CHAPTER 5

Experiments Conducted on a Simple Brass Instrument

5.1 Introduction

In the previous Chapter, results were presented which confirmed the suitability of the LDV technique, and the basic experimental set-up, for measuring the wall vibrations induced when a tube is excited by a method. In this Chapter, experiments designed to study the wall vibrations induced when a simple brass instrument is blown are described. These experiments involve measuring both the instrument's air column resonances and its structural resonances and then comparing the results with measurements of the velocity amplitude variation along the pipe wall when the instrument is blown. The simple instrument consisted of a cylindrical acoustic resonator (metal pipe) and a trombone mouthpiece and was used in this basic format throughout the thesis. The air column was set into resonance using an artificial blowing mechanism, which will be described in detail later.

5.2 Experimental Brass Instrument

The simple instrument used at this stage of the research comprised a straight length of brass pipe (length 70 cm, outside radius 0.7 cm and wall thickness 0.5 mm) coupled to a Denis Wick trombone mouthpiece. The instrument can be classified as an aerophone; that is, an instrument where a vibrating column of air produces the sound. It can be further described as a lip-reed instrument, the buzzing of a player's lips within a mouthpiece producing the air pressure fluctuations necessary to sound the instrument. It is therefore acoustically similar to the instruments of an orchestral brass section.

Where this simple brass instrument differs from its orchestral counterparts is that its bore is cylindrical for its entire length and has no bell flare. With the more widely spaced resonances that this profile produces, its musical effectiveness is reduced. As was discussed in Chapter 2, an orchestral brass instrument has varying ratios of conical and cylindrical lengths terminating in a bell flare and as a result its resonances are nearly harmonically related. The lack of the bell in the simple instrument also reduces the degree of coupling to the surrounding environment, and consequently the instrument's attainable volume is much less than that of orchestral instruments.

To increase the musicality of the simple instrument, the fashioning of a single bell flare that could be transferred between each pipe under test was considered. There were three main reasons for the rejection of this addition to the test object:

- The harmonicity of the resonances was not an important consideration in this study. There was no musical sequence to be played, measurements were all to be made at one single pitch.
- Sound power output was also not an issue, as measurements would be conducted at a set distance within an anechoic chamber.
- It would create an extra variable in the experiment whose clamping conditions would have to be redesigned to accommodate it.

5.3 Artificial Blowing Mechanism

To take a series of vibrational measurements along the length of an instrument with the objective of later making a comparison of wall vibration between different pipes, a note has to be maintained for up to thirty minutes (an hour with the initial Dantec based measurement system). It would be impossible for a musician to maintain a note at a steady sound pressure level for even a small percentage of this time. A musician would also be unable to reproduce the note exactly, if a measurement had to be taken repeatedly at one pitch.

Consequently, an artificial mouth was constructed which would sufficiently mimic a human player to allow single tones to be played over long periods. This would also remove the problem of any variables introduced by a brass player, such as an increase in the speed of sound due to breath warming the air column. Figure 5-1 is a photograph of the artificial mouth used in the study. The blueprint for the artificial mouth used in the experimental set-up was a version of an early design supplied by the Acoustics and Fluid Dynamics Group at Edinburgh University and based on the original design of Joël Gilbert and Jean-François Petiot [1998].

The artificial mouth consists of a hermetically sealed hollow Perspex box (internal dimensions 14 cm x 10 cm x 9 cm) representing the mouth cavity and vocal tract. Figure 5-2 shows a schematic of the mouth in vertical cross section. The walls are made of individual perspex plates with the front plate being detachable to allow access to the mouth's interior. The side plates are relatively thick giving an increased area of contact with the front plate and thus providing a good airtight seal. A Denis Wick trombone mouthpiece is permanently fixed into the front plate. The plate was fixed in position using screws, with vacuum grease applied to the interfacing surface to provide an airtight seal.

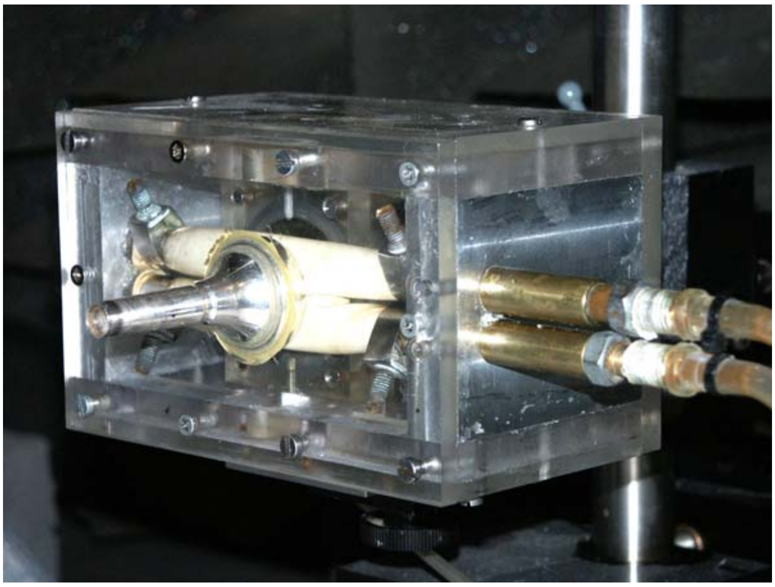


Figure 5-1. The artificial mouth used in the experimental set-up

A pair of thin, water filled, cylindrical latex rubber tubes (internal diameter 15 mm, wall thickness 0.2 mm) are positioned parallel to each other within the mouth in such a way that, when the front plate is fitted, the mouthpiece almost touches the lips and approximately covers each lip equally. The latex tubes attempt to mimic, the density and flexibility of musicians' lips.

The end of each latex tube is fitted over the end of a hollow support rod and held in place with a jubilee clip. The support rods on the right side of the artificial mouth are fixed in the side plate. The support rods on the left side of the artificial mouth are moveable, having an outside thread which interfaces with an internal screw thread of metal tubes which are themselves fixed in the side plates of the mouth. This forms a lip tension control mechanism. Rotation of this mechanism displaces the moveable lip support rod in a longitudinal direction and in the process adjusts the longitudinal tension of the lips.

Behind the lips a rectangular perspex block, with a hole of slightly narrower diameter (approximately 2 cm) than the mouthpiece drilled through its centre, simulates the effect of a player's teeth. The teeth are supported by an annular rim on a support tube and guided by pins fitted into grooves in the top and bottom plates. The annular rim can rotate freely within an annular channel at the rear of the teeth but the rim and channel cannot be separated. An outer screw thread at the rear of the teeth support tube interfaces with an inner screw thread of the teeth control mechanism, which protrudes through the rear of the artificial mouth. Rotation of the teeth control mechanism displaces the teeth support tube, and hence the teeth, in a transverse (front - rear) direction. In this way, the mechanical pressure of the lips against the mouthpiece can be varied. Large sections of the teeth support tube have been removed to ensure a uniform pressure field exists across the rear face of the lips.

A brass player forms their embouchure by tensing the corners of the mouth and buzzing the lips while pressing the mouthpiece against the lips. The tongue moves back and forth behind the teeth, striking the area along the top of the back of the upper teeth for the creation of each separate note (known as the 'attack').

In the artificial approximation, the tension of the latex lips is preset when they are fixed within the artificial mouth. Tension is placed upon the lips by adjusting the teeth, pressing them into the mouthpiece while adjusting the lip volume as necessary by addition /removal of water (via plastic tubing and valve attached to the lips through the side plate of the mouth). It was hoped that the tension could be adjusted by external manipulation of the support rods but this proved very problematic due to air leakage, any leakage from the mouth made sustaining a note almost impossible. This meant there was a restriction on the degree of manipulation, making the range of available notes less easily achievable.

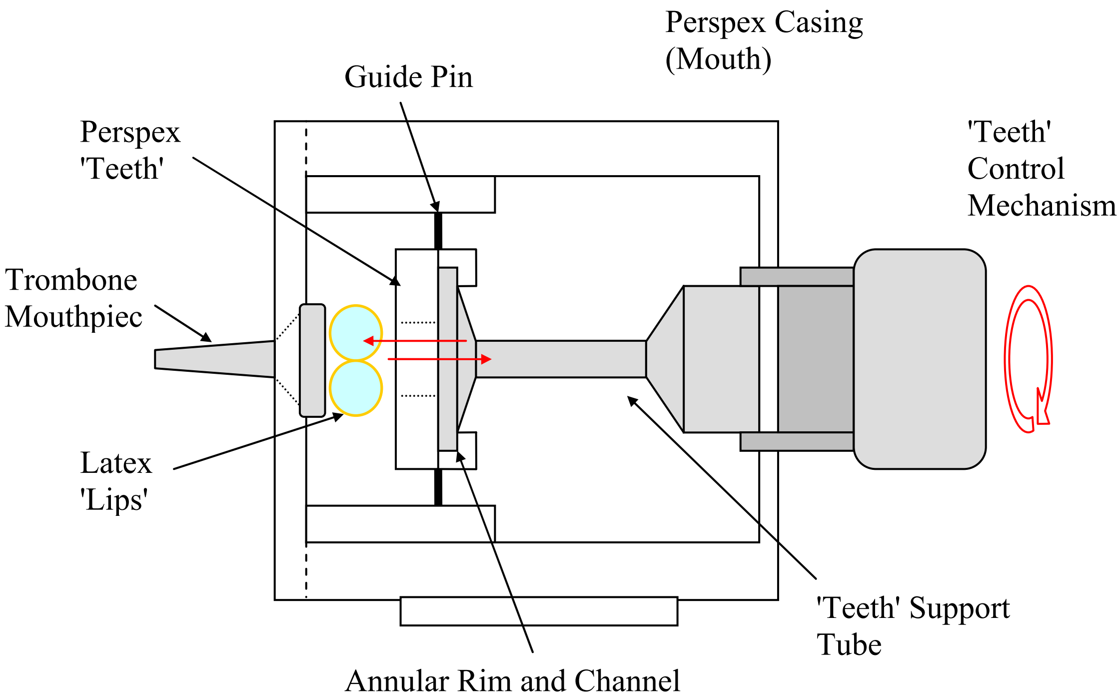


Figure 5-2. A cross section of the artificial mouth used in the experimental set-up

A constant air supply, representing air from the lungs, is required to maintain a static mouth overpressure and so a stable note. This constant flow of air is provided by a Vane generator and valve system which is connected to an air inlet port in the side plate of the artificial mouth. The air flows through the teeth and between the lips.

Overpressure within the mouth is monitored by a Furness Controls Ltd. FC014 micromanometer connected via a narrow flexible tube to an air outlet port in the side plate of the artificial mouth. The micromanometer was also used to check the suitability of the air supply; with the mouthpiece blocked, the resultant steady micromanometer reading confirmed that no fluctuations in the air pressure were present.

The simple construction of this artificial mouth was preferred to later, more advanced, versions. This was due to the lips being sealed within the mouth rather than fitted externally as they have been in later models. These later models give much easier access, and greater flexibility of adjustment, to the lips but would be less likely to provide the total air leakage free state required for this series of experiments.

5.4 Input Impedance of the Simple Brass Instrument

5.4.1 Introduction

The input impedance Z of a musical wind instrument is a measure of the amplitudes and frequencies of the instrument's resonances. It is defined as being the ratio of acoustic pressure p to volume velocity U at the entrance of the instrument:

$$Z = \frac{p}{U} \tag{5.1}$$

To determine the resonance frequencies of the simple brass instrument, its input impedance was measured using a conventional frequency domain (swept sine wave) technique.

5.4.2 Experimental Set-up and Procedure

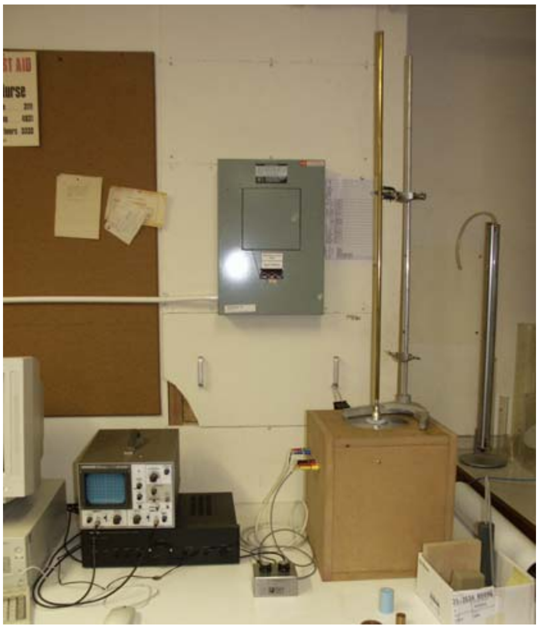


Figure 5-3. Input impedance experimental set-up

Figure 5-3 shows a photograph of the experimental set-up used to measure the input impedance of the simple brass instrument. A loudspeaker, housed within a wooden casing, is coupled to an annular capillary.

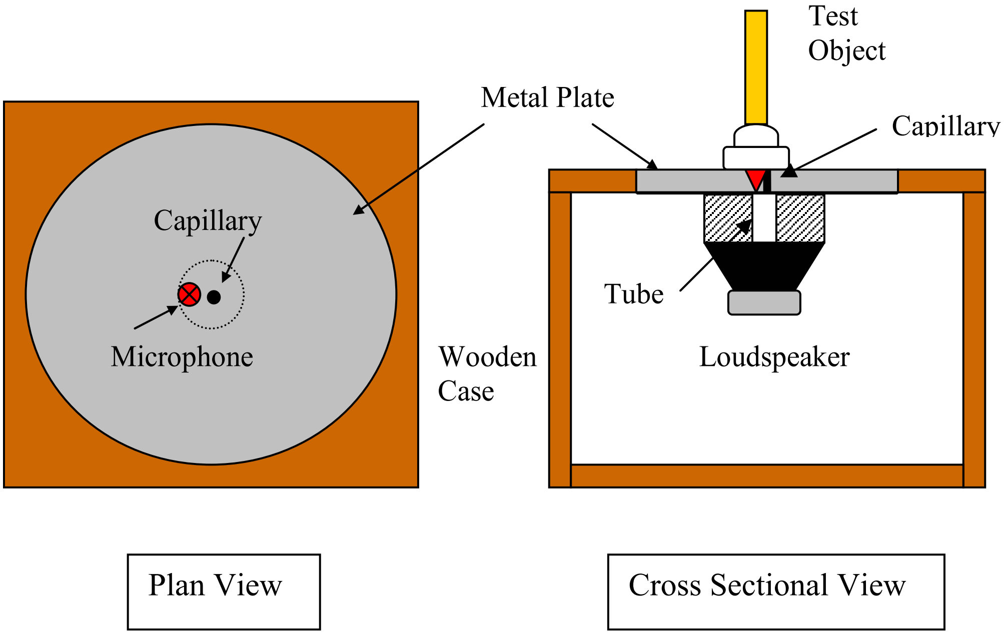


Figure 5-4. Diagram of experimental set up for impedance measurement.

The capillary is set within a metal plate which is built into the top of the wooden casing. A response microphone is embedded in the plate in close proximity to the capillary (Figure 5-4).

The instrument to be measured is positioned over both the capillary and the microphone and the connection is sealed to prevent air leakage. A computer-based measurement system drives the loudspeaker at a particular frequency, thus feeding a sinusoidally varying pressure wave via the high impedance capillary into the mouthpiece of the instrument. The pressure response is recorded by the microphone. This is repeated for each frequency of interest.

On the assumption that the capillary impedance is independent of frequency and much larger than the instrument's impedance, the excitation wave can be considered to have a constant volume velocity. The recorded pressure response is then directly proportional to the input impedance of the instrument. Thus, the impedance of the instrument can be found by dividing the pressure response by the volume velocity. (The volume velocity is determined by performing a calibration measurement on a small volume of known impedance).

5.4.3 Results

Figure 5-5 shows the input impedance of the simple brass instrument, measured over a frequency range of 50 Hz to 2000 Hz with a 1 Hz resolution. The input impedance of the mouthpiece alone has also been superimposed on the graph.

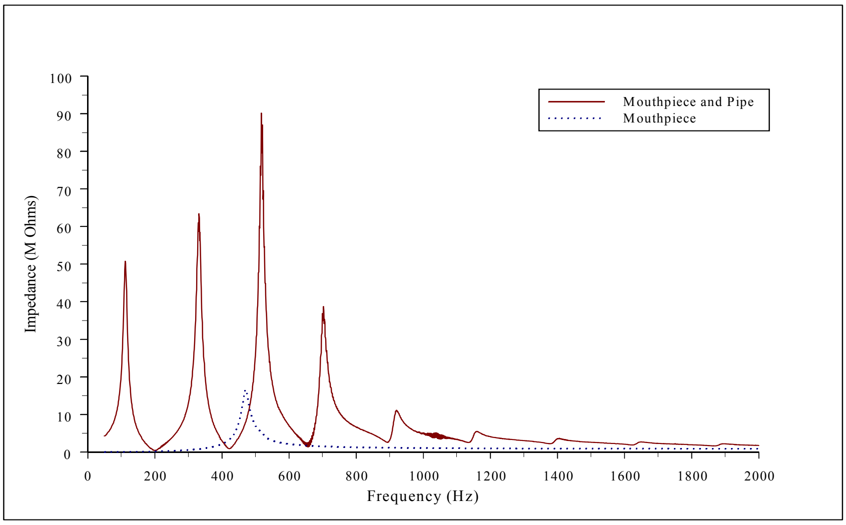


Figure 5-5. Swept sine wave response curve for the brass pipe coupled to a trombone mouthpiece and for the mouthpiece alone

The peaks in the impedance curve for the brass pipe coupled to the mouthpiece occur at the resonance frequencies of the instrument. They therefore give an indication of the pitches that the instrument will produce. In this case, the peaks occur at 112 Hz, 332 Hz, 518 Hz, 703 Hz and 921 Hz so one would expect the instrument to be capable of producing notes with these as their fundamental frequencies.

The impedance peaks increase in amplitude until reaching a maximum at the 3rd resonance. This may be attributed to approaching the mouthpiece resonance frequency of 472 Hz. Above the 3rd resonance, the peaks begin to diminish in amplitude again. This corresponds to the wave energy beginning to be radiated away rather than being reflected and contributing to standing wave production.

The amplitude of the mouthpiece impedance curve is lower than expected. Most likely due to difficulties with the experimental apparatus this, was not of great concern for this study as it is the frequency at which the peak occurs that is of interest.

5.5 Determination of Structural Modes for the Brass Instrument

5.5.1 Introduction

In this section, the structural modes of the simple brass instrument are measured under two different clamping conditions. In the first condition, both ends of the instrument are rigidly clamped using jubilee clips. In the second condition, the mouthpiece end is rigidly clamped while the open end of the pipe is loosely suspended by a loop of cotton and allowed to vibrate freely.

5.5.2 Experimental Method

The brass pipe was rigidly clamped by a jubilee clip at each end around the entire circumference to prevent movement perpendicular to the length of the pipe. A cushion of foam was attached to the inner surface of the jubilee clips to ensure a firm contact around the entire circumference. The pipe was fixed horizontally on an anti-vibration optic table housed in a semi-anechoic chamber. Although it was not activated for this first series of measurements, the mouthpiece of the artificial mouth was inserted into one end of the pipe (this was to ensure that the structural modes measured were for the whole pipe/artificial mouth combination, and consequently they could be compared with later measurements of the velocity amplitude variation along the pipe when the whole instrument is blown).

In order to excite the mechanical resonances, the pipe was driven at a position close to the mouthpiece using a Ling Dynamic Systems V200 shaker with a fine pointed needle attachment. The vibrometer laser beam was reflected and focussed on to the top of the pipe using a silver-sided mirror angled at 45º to the light path. To determine the mechanical resonances the shaker was driven (using a Thurlby Thandar Instruments (TTi) TGA 1230 synthesised arbitrary waveform generator connected to a Brüel & Kjaer type 2706 power amplifier) at discrete frequencies over a range of 20 Hz - 1 kHz in 10 Hz steps, and at each frequency the velocity amplitude (in m s^{-1}) was measured using the Dantec LDV. The upper frequency limit was chosen to be 1 kHz as the impedance curve for the instrument (Figure 5-5) indicates that the major air column resonances occur below this frequency. Readings were taken along the pipe at 6 cm intervals.

The clamp was removed from the open end of the pipe and the procedure repeated with the end loosely suspended in a loop of cotton.

Ling Dynamic Systems V200 shaker

The Ling Dynamic Systems V200 series electro-dynamic vibration unit or 'shaker' has a height 96 mm, diameter 78 mm, weight 1.81 kg and a useful frequency range of 5–13000 Hz. It functions by the interaction between a steady magnetic field, produced by a permanent magnet, and an oscillating current flowing in the moving coil. A force is generated at right angle to the lines of flux and to the conductor carrying the current which is proportional to the product of the instantaneous current and the magnetic flux density.

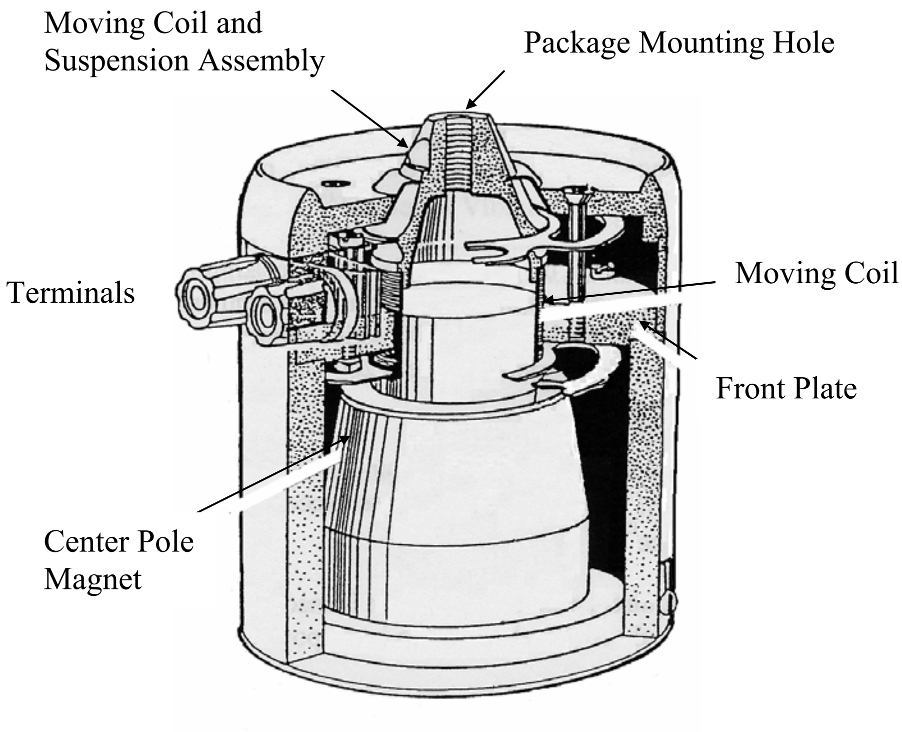


Figure 5-6. V200 series vibrator (shaker) - Sectioned view

5.5.3 Results

The results of the artificial mouth/brass pipe combination wall response while being mechanically shaken can be seen in Figures 5-7 and 5-8.

Figure 5-7(a) shows at 2D contour plot of the variation in velocity amplitude with frequency along the length of the instrument with both ends clamped. The natural vibrational modes are clearly distinguishable in the plot as are the frequencies of their occurrence, the first four modes (bending modes) being located at 80 Hz, 230 Hz, 500 Hz and 830 Hz. The relative amplitudes of the modes can be more easily made out in Figure 5-7(b), which shows a 3-D representation of the plot.

For comparison purposes, theoretical predictions of the first four bending mode frequencies (55 Hz, 220 Hz, 495 Hz and 880 Hz respectively) have been superimposed on Figure 5-7(a). These values were calculated from the formulae presented in Section 4-4 using the following parameters for the brass pipe:

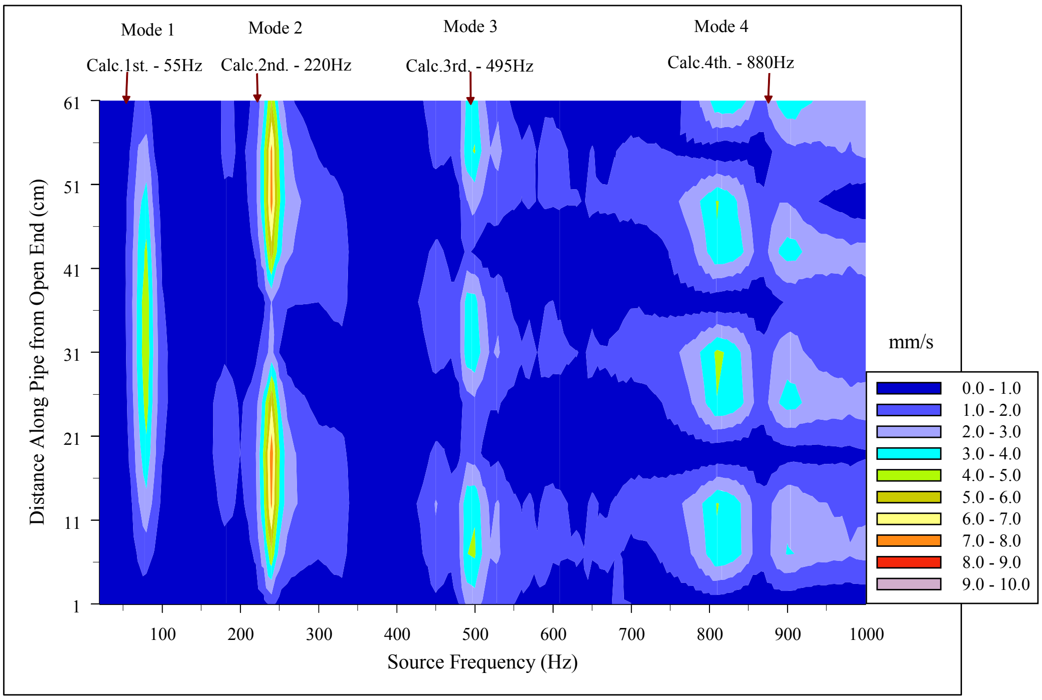
ρ - Density of cylinder materials = 8400 kg/m³ (ρ, E, ν)[1]
E - Modulus of elasticity = 106 GPa
ν - Poisson's ratio = 0.34
R - Cylinder radius to midsurface = 0.0065 m
h - Cylinder thickness = 0.000535 m
L - Length of cylinder = 0.683 m

It should be noted that brass is not a single unique metal but rather a series of copper based alloys in which the other main component is zinc (the relative percentages of the two metals depend on the type of brass). As a result, the physical properties such as density and modulus of elasticity vary between brass types.

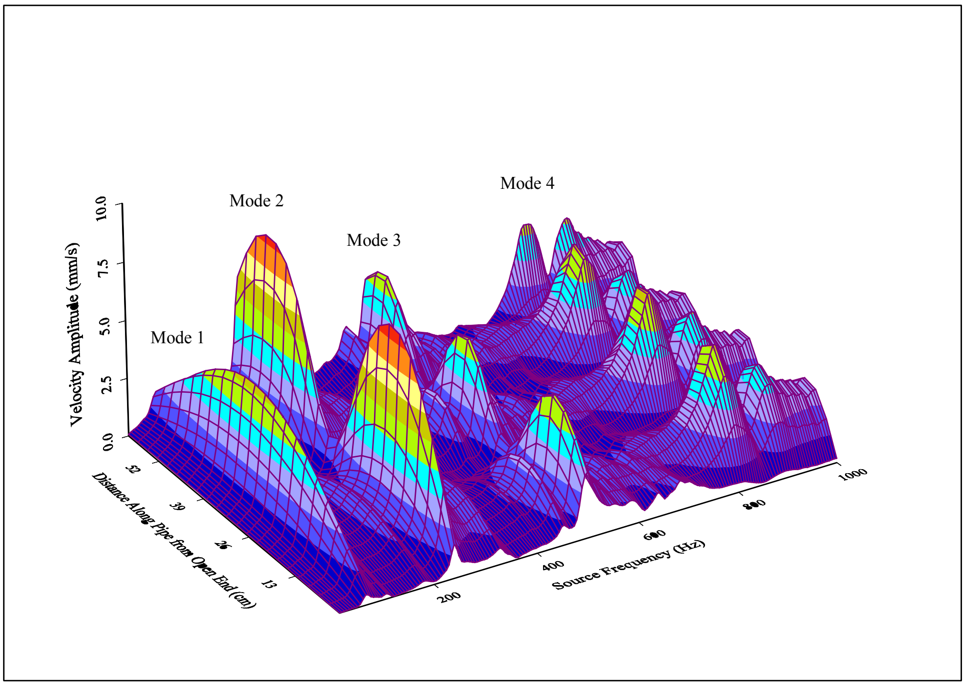
It should also be noted that the formulae of Section 4-4 revealed that, for this brass pipe, the other structural mode types (e.g. radial modes, axial modes etc) occur at much higher frequencies than the than the 20 Hz – 1 kHz range investigated in these experiments.

Figures 5-8(a) and (b) show a 2D contour plot and 3D plot for the brass pipe measured with the open end of the pipe loosely suspended by a loop of cotton. Once again the plot clearly shows distinct modal patterns at the natural frequencies of vibration but the first four bending modes have been shifted in frequency to approximately 30 Hz, 130 Hz,
300 Hz and 640 Hz. The mode shapes have also changed, as the open end is now able to vibrate freely.

[1] Engineering Fundamentals: Properties of common solid materials. http://www.efunda.com/materials

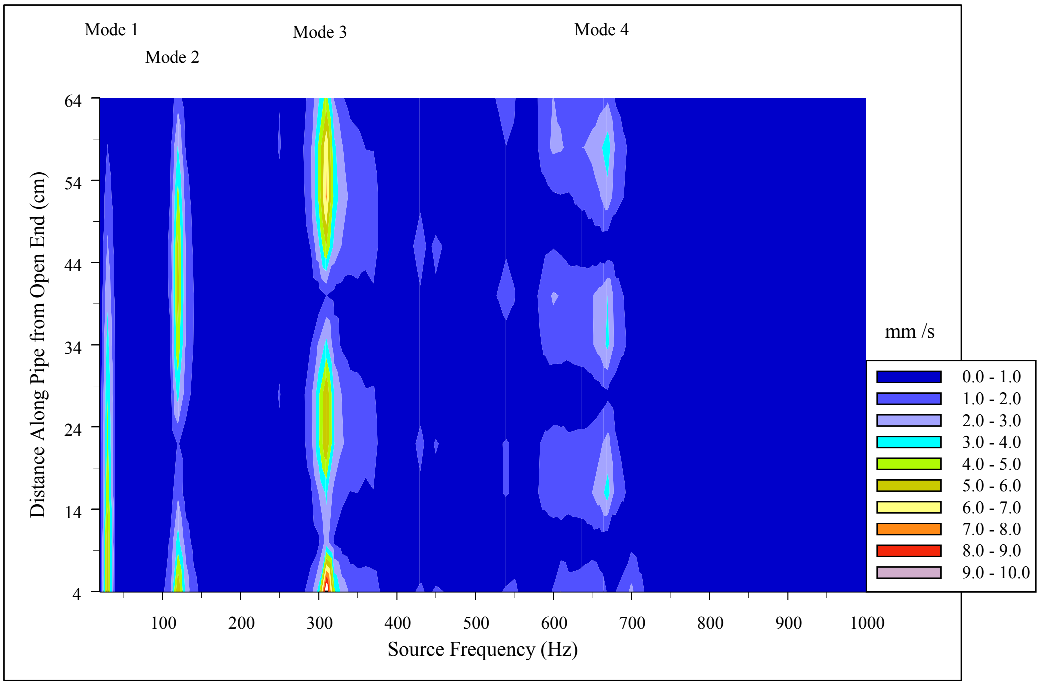


(a)

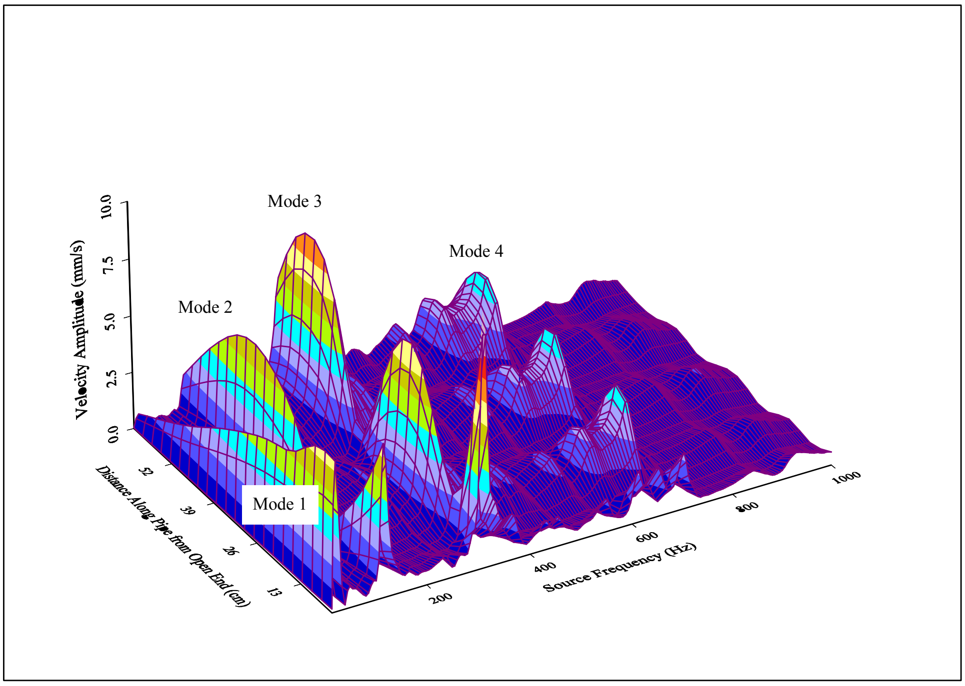


(b)

Figure 5-7. (a) 2D contour plot and (b) 3D plot of velocity amplitude variation with frequency along the pipe with both ends clamped



(a)



(b)

Figure 5-8. (a) 2D contour plot and (b) 3D plot of velocity amplitude variation with frequency along the pipe with one end loosely suspended

5.6 Determination of Operational Deflection Shapes

5.6.1 Experimental Method

The shaker was removed from the set-up described in Section 5.5.2 and the air-supply to the artificial mouth activated. The lips were adjusted until a discernible stable note was heard in a selected frequency region. This note was recorded using a Brüel & Kjær half inch microphone and Power Amplifier Type 2706. The microphone was positioned 1 metre from the open end of the instrument. Frequency analysis revealed a fundamental frequency of 333 Hz, a second harmonic at 666 Hz and a third at 1000 Hz. The velocity amplitudes at 3 cm intervals along the pipe were measured using the LDV. The set up can be seen in Figure 5-9.

This procedure was performed with both ends of the pipe clamped and then repeated with the far end loosely suspended. The pressure and embouchure were maintained at a constant level throughout to allow the amplitudes of vibration to be compared.

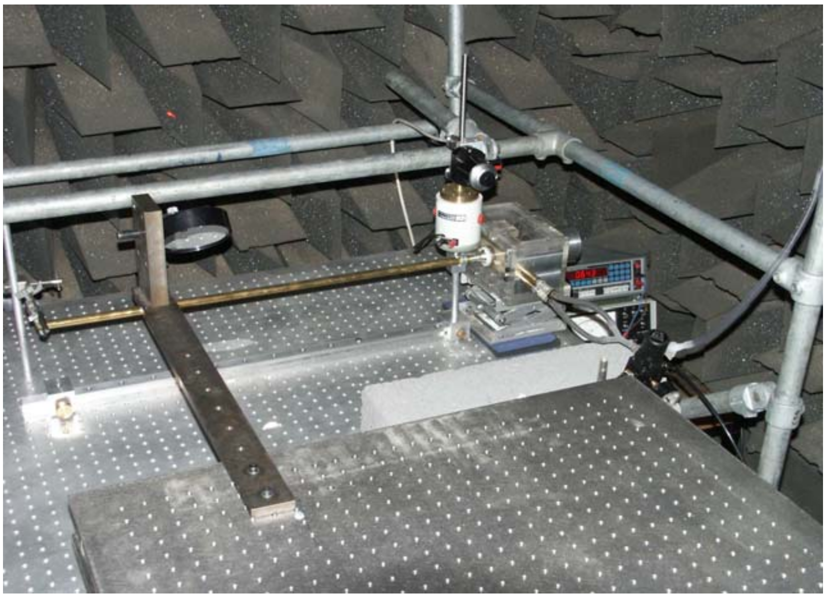
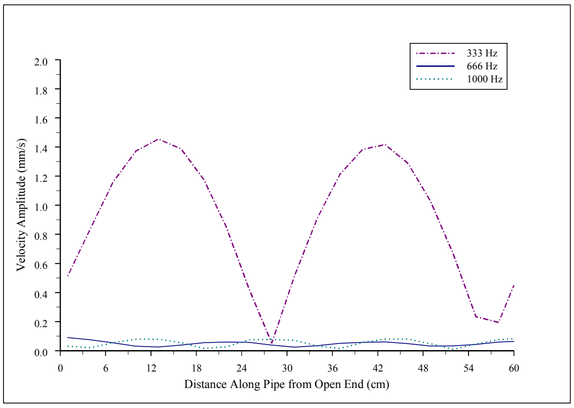


Figure 5-9. Experimental set up of the simple brass wind instrument

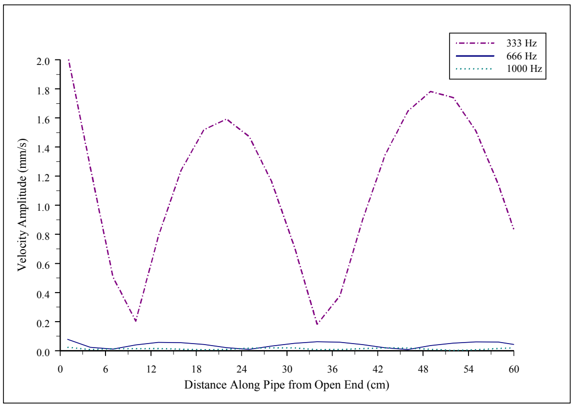
5.6.2 *Induced Wall Vibration Results*

An FFT of the signal measured by the LDV showed the vibration at each point was composed of three separate frequencies.

Figure 5-10(a) shows the velocity amplitude variation along the length of the pipe induced by the artificial mouth plotted at these frequencies (333 Hz, 666 Hz and 1000 Hz.) when both ends are clamped. Comparison with the 2D contour plot for the instrument in the same clamping configuration (Figure 5-7(a)) shows that the variation in the velocity amplitude at the fundamental frequency of the played note (333 Hz) matches the natural bending mode shape centred on 230 Hz. Similarly, the plot for the second harmonic of the played note (666 Hz) corresponds to the natural bending mode shape centred on 500 Hz and the third harmonic (1000 Hz) corresponds to the natural bending mode shape centred on 830 Hz. The second and third velocity amplitude variation can be more easily seen in Figure 5-11(b) and (c) respectively.



(a)



(b)

Figure 5-10. Velocity amplitude variation along the pipe at 333 Hz, 666 Hz and 1000 Hz when artificially blown with (a) both ends clamped and (b) loosely suspended

Figure 5-10(b) shows the velocity amplitude variations induced by the artificial mouth at the same three frequencies under the second clamping condition of a loosely suspended open end. Comparison with the 2D contour plot for the instrument in the same clamping configuration (Figure 5-8(a)) again shows a match to the bending mode shapes of the pipe. However, in this case the two harmonics of the played note are closer in frequency to the structural resonances of the pipe.

There is an importance in the agreement between the harmonic frequencies of the played note and the frequencies of the instrument's structural modes to the amplitude of the induced wall vibrations.

Figure 5-11(a) shows a comparison of the velocity amplitude variations along the pipe induced by the artificial mouth at 333 Hz for the pipe clamped at each end compared to when the open end is freely suspended. The peak velocity amplitude measurements are slightly larger when the pipe is loosely suspended at one end. This is consistent with Figures 5-7(a) and 5-8(a) which show that when the pipe is clamped at both ends the closest structural resonance to 333 Hz is the bending mode centred on 230 Hz. When the pipe is suspended at one end the closest structural resonance to 333 Hz is the bending mode centred on 310 Hz.

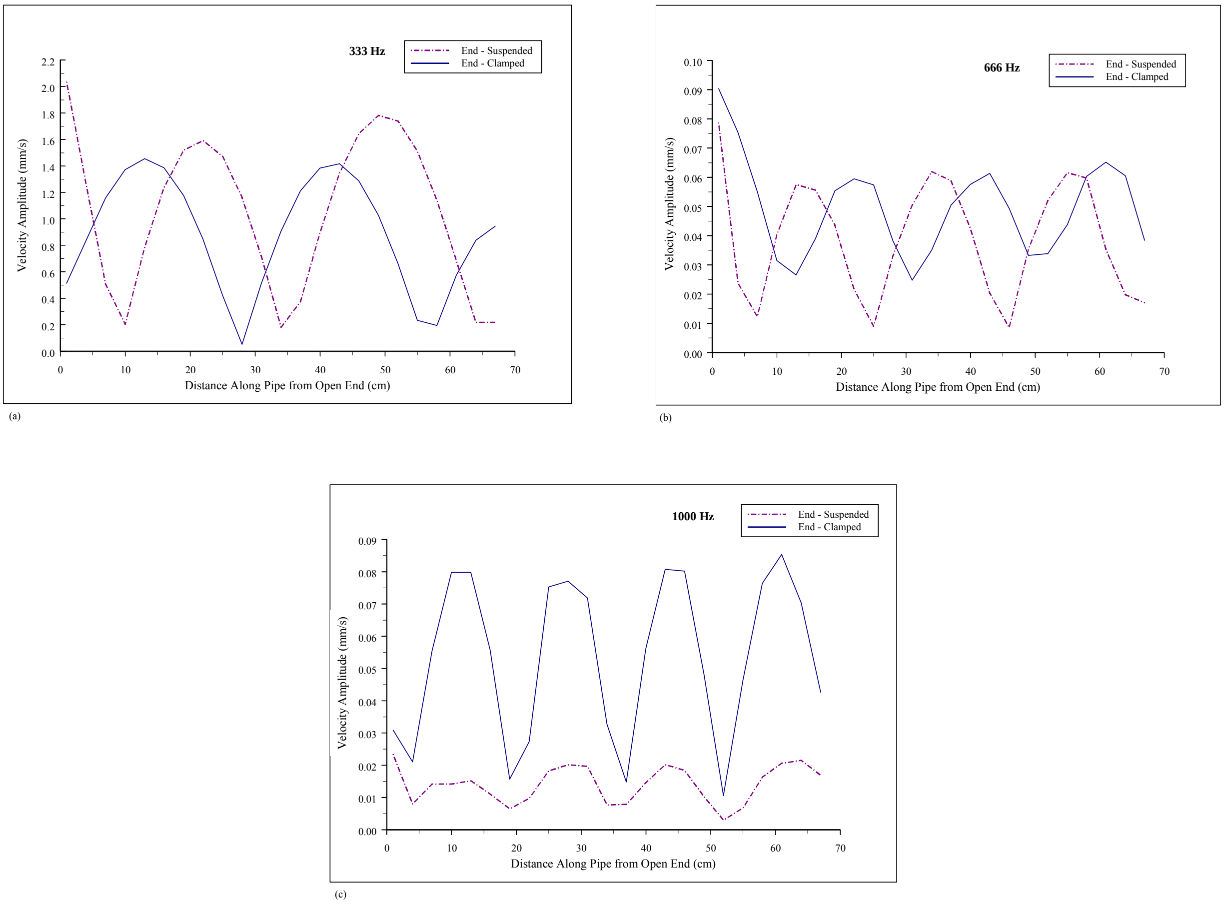


Figure 5-11. Velocity amplitude response for a pipe with a loosely suspended and clamped open end at (a) 333 Hz (b) 666 Hz and (c) 1000 Hz

Figure 5-11(b) compares the velocity amplitude variations along the pipe induced by the artificial mouth at 666 Hz under the two clamping conditions. In this case, at the antinodes, the velocities are approximately the same whether the open end of the pipe is rigidly clamped or loosely suspended. However, at the nodes, the velocities are lower when the pipe is loosely suspended. Again, this is consistent with Figures 5-7(a) and 5-8(a) which show that at 666 Hz, although the structural responses under the two clamping conditions are similar, in the loosely suspended case there is a well defined bending mode centred close by which may explain the deeper troughs.

Figures 5-11(c) compares the velocity amplitude variations along the pipe at 1000 Hz under the two clamping conditions. Here, at the antinodes, the velocities are significantly larger when the pipe is clamped at both ends. Examination of Figures 5-7(a) and 5-8(a) reveals that under rigid clamping conditions the effect of the fourth bending mode is still evident at 1000 Hz. When the pipe is loosely suspended at one end the structural response at 1000 Hz is much lower.

5.7 Comparison of Artificial Mouth with Human Player

To ascertain that velocities induced in the wall of the instrument when artificially blown are comparable to those produced by a musician when playing a particular note, a player attempted to attain a note of a similar pitch and loudness to that used in the artificially blown experiments.

5.7.1 Experimental Method

LDV velocity measurements were undertaken on the artificially blown instrument as described previously (Section 5.5.2). As results obtained from a human player were to be compared to these artificially blown measurements, the instrument was firmly clamped at each end. A loosely suspended open ended pipe had too much lateral movement generated by the coupling with the musician, a small degree of movement making LDV measurements impossible as the laser beam will be reflected off the curved surface of the pipe away from the sensor head. At each measurement point the sound pressure level was found using a Brüel & Kjær half inch microphone and Power Amplifier Type 2706. The microphone was positioned 1 metre from the open end of the instrument, the level in dB noted on a Brüel & Kjær measuring amplifier Type 2610.

A musician entered the chamber and was invited to blow the instrument, attempting to produce a note of similar fundamental frequency, and with a sound pressure level of less than 2 dB difference from that produced when artificially blown. When a suitable note was achieved the musician was asked to repeat this for each data point while measurements were recorded with the LDV.

The instrument had to be sounded while fixed to the anti-vibration table. The cramped and awkward posture the player was forced to adopt meant repeated attempts had to be made to produce a steady note with the correct measurement criteria for a sufficient period of time. The equipment was monitored and a 'snapshot' measurement taken when such a note was achieved.

5.7.2 Experimental Results

Figure 5-12(a) shows the velocity amplitude variation along the pipe induced when blown by a human player. For comparison purposes, Figure 5-12(b) shows the variation induced by the artificial mouth. In both cases, the velocity variations at the fundamental and second harmonic of the played note are shown.

The two graphs show good agreement in shape and amplitude, especially at the fundamental frequency of the played note. The second harmonic is much more prominent when produced by a musician compared with the more pure tone of the artificial mouth. This confirms the acceptability of using the artificial mouth when measuring the wall vibrations induced when an instrument is blown.

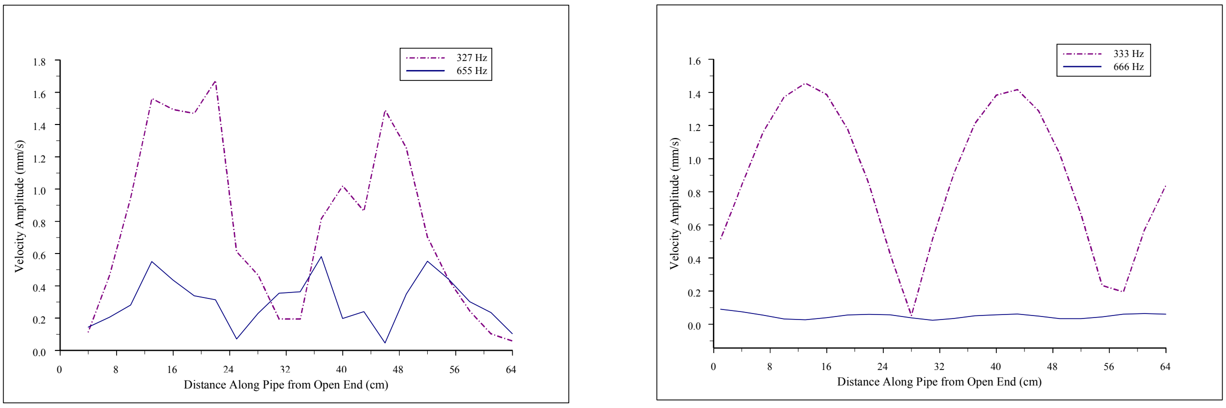


Figure 5-12(a) Velocity amplitude variation produced by a human player. (b) Velocity amplitude variation produced by artificial mouth under similar conditions

The difficulty that the player had in producing notes of similar loudness and quality is evident when Figures 5-12(a) and 5-12(b) are compared. The constant output of the artificial mouth results in the much smoother variation in velocity amplitude along the pipe seen in Figure 5-12(b). The artificial mouth removes the problems encountered when using a human player, such as the introduction of unavoidable movement into the system and the inability to repeat and sustain notes over long periods.

5.8 Problems Encountered with Artificial Mouth Operation

The maintaining of a note at a constant pressure and frequency proved more problematic than was originally envisaged, the main problem encountered being the prevention of any air leakage from the artificial mouth.

The leakage of any air from the control mechanisms of the artificial mouth or from between the perspex plates from which it is constructed would make maintaining a note of constant frequency and pressure practically impossible. Unfortunately, leakage was an ongoing problem throughout the study causing frequent difficulties in the taking of complete and accurate sets of measurements.

In an attempt to alleviate the problem the longitudinal tensional control mechanism was permanently locked into position and the thread sealed with PTFE tape. Lip tension was thus limited to that preset when the latex tubes were fitted within the mouth. The only control available over tension was the pressure exerted on the lips by the teeth moving against the mouthpiece coupled to the variation in water volume within the lips. The limitation on lip flexibility was not such a problem in the intended study as only a small number of different notes were needed, a constant pitch produced over a reasonably lengthy period being by far the more important criteria. When adjustment of the lips was first made on activation of the air supply, care had to be taken with teeth and air pressure as the lips could easily be blown shut in the mouthpiece. This effectively sealed the artificial mouth and could destroy the airtight seal resulting in air leakage.

Furthermore, after a note was obtained at a desired pitch, a certain degree of change in the fundamental frequency could occur until the lips 'settled' producing a steady note. Even then there is a degree of ‘creep’ in the lips over long periods of time encouraging measurements of the blown instrument to be concluded as quickly as possible.

5.9 Conclusion

Experiments have shown that when a simple wind instrument, consisting of a mouthpiece and section of brass piping, is artificially blown mechanical wall resonances are excited. The strength of these induced wall vibrations is dependent on how close in frequency the air column resonances and the structural resonances are. As clamping conditions significantly affect the structural resonance frequencies, this will have to be taken into consideration when comparing resonators of different materials.

The artificial mouth has been shown to be a more than suitable substitute for a human player when measuring induced wall vibrations. The ability of the artificial mouth to repeat and sustain notes at a given loudness should make it an excellent tool when comparing the vibrations induced in pipes of different materials and wall thickness.

CHAPTER 6

Investigation of Excitation Mechanism

6.1 Introduction

The experiments described in Chapter 5 have shown that using an artificial mouth to blow a simple brass instrument excites its wall into vibration at frequencies that match those of the air column resonances. These wall vibrations could be caused through different mechanical processes.

When a brass instrument is blown, the strong coupling between the air column and the lips means the lips end up oscillating at, or close to, the resonance frequencies of the air column (this is demonstrated in Section 6.2 where a photodiode is used to determine the frequency of vibration of the lips). Therefore, the vibrations in the wall of the instrument may be a response to the oscillation of the lips which are in contact with the pipe through the mouthpiece, the lips effectively acting as a mechanical shaker. However, the vibrations could also be a result of coupling between the air column resonances and the structural resonances. That is, the air pressure changes within the pipe might be providing a driving force to excite the wall into vibration. Either way, the wall vibrations are only likely to be significant when the air column resonances are close in frequency to the structural modes.

Two experiments were carried out to help determine the dominant mechanism by which wall vibrations are excited. The first (described in Section 6.3) was designed to decouple the lips from the walls of the instrument. This involved inserting a short length of flexible tubing between the mouthpiece and the brass pipe. The second (described in Section 6.4) was designed to decouple the air column from the walls of the instrument. This involved inserting a tube of narrower diameter within the brass pipe.

6.2 Photodiode Measurements of Lip Motion

The interrelationship of the lip, air resonance and wall vibration frequencies induced when a note is played using the artificial mouth was observed by performing photodiode measurements of the lip motion, microphone measurements to establish the frequency components of the note produced, and LDV measurements of the pipe wall motion. The sound pressure recordings were taken using a Brüel & Kjær half inch condenser microphone (cartridge type 4133) and a Brüel & Kjær Dual Microphone Supply, Type 5935.

6.2.1 Experimental Method

A section of brass pipe (length 70 cm, outside diameter 14 mm and wall thickness 0.5 mm) was used as the instrument's acoustic resonator for this series of experiments and set-up as for the artificially blown vibrational measurement of a simple instrument previously undertaken in Chapter 5.

The artificial mouth was coupled to the pipe via the mouthpiece, the air supply activated, and the instrument artificially blown at a particular frequency. The Dantec LDV system was used to obtain the peak velocity amplitude variation along the pipe measured at twenty three points at 3 cm intervals.

A small, shielded light source was placed to the rear of the artificial mouth, a glass plate in the teeth adjustment control allowing the passage of light through the mouth, lips, and along the bore of the instrument. All other sources of light in the anechoic chamber were extinguished and measurements taken.

The interruption in the passage of light caused by the movement of the lips was recorded as changes in voltage by the photodiode, allowing the lip frequency to be determined. The sound pressure level of the blown note was recorded at the same time using the condenser microphone. Both were measured over a 10 second period and saved as .wav sound files to be later processed using the frequency analysis software, Sigview.[2]

6.2.2 Experimental Results

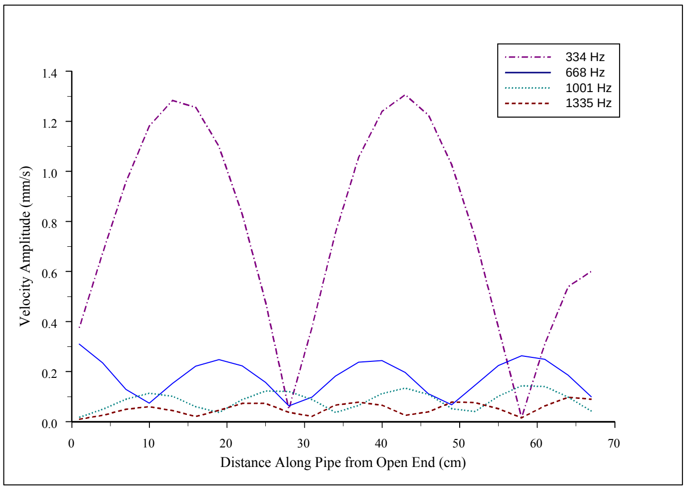
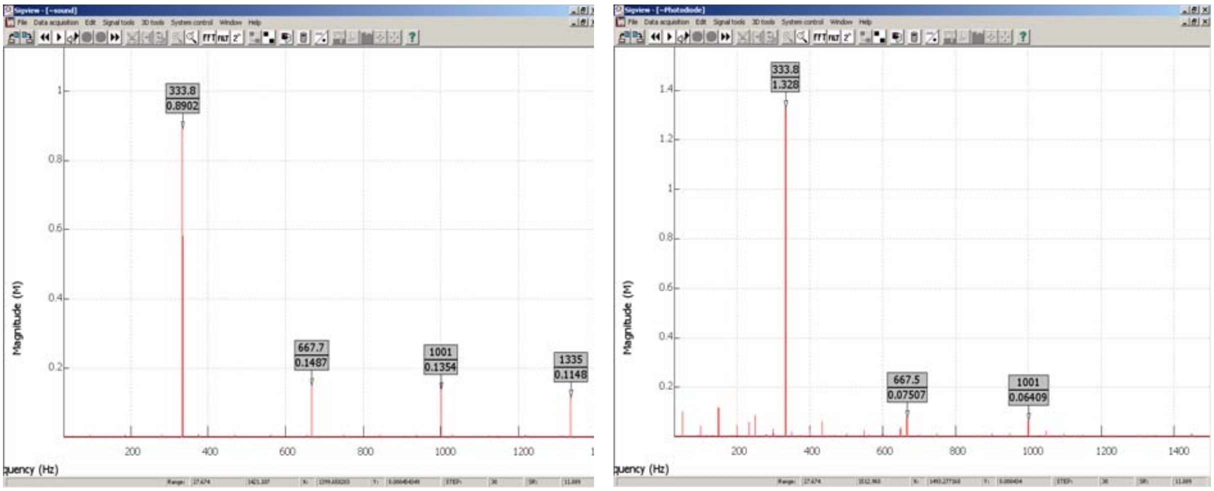


Figure 6-1. Velocity amplitude variation of pipe measured at the first four harmonics 334 Hz, 668 Hz, 1001 Hz and 1335 Hz

Figure 6-1 shows the variation in vibrational velocity along the length of the pipe for the first four harmonics, 333.8 Hz, 667.7 Hz, 1001.6 Hz and 1335.9 Hz, the first mode being

[2] FFT Signal analysis software. http://www.sigview.com/.

by far the dominant vibrational velocity of the pipe wall. The shapes of each mode can be clearly distinguished.



(a) (b)

Figure 6-2. Linearly plotted frequency spectrum for (a) sound pressure recording and (b) photodiode signal

The voltage variations measured by the microphone (sound pressure) and by the photodiode were both approximately sinusoidal in nature, indicating one dominant frequency. This was confirmed by performing an FFT on each measurement to produce its frequency spectrum. Figure 6-2(a) shows a frequency spectrum of the recording of the tone produced by the instrument and reveals a fundamental frequency of 333.8 Hz with higher harmonics at 667.7 Hz, 1001 Hz and 1335 Hz. Figure 6-2(b) shows the frequency spectrum of the variation in light intensity through the instrument caused by the opening and closing of the lips. The spectrum is dominated by a frequency of 333.8 Hz. Although there is interference from mains hum the higher harmonics of 667.5 Hz and 1001.3 Hz that were also present in the sound pressure FFT can still be picked out.

The lip and sound pressure frequencies both match the vibrational frequencies seen in the pipe walls. The note obtained on the instrument was playing at a fundamental dominant frequency of 334 Hz, which coincides with the second resonant frequency of the instrument of 332 Hz found by input impedance measurement (Figure 5-5). It needs to be seen whether it is the air resonances or motion from the lips directly which exerts the most influence over the pipe wall.

N.B. In the experiments described in the previous chapters, the mouthpiece end of the pipe was concealed by the mechanical shaker and was therefore unable to be scanned by the LDV. With the shaker now removed, this region of the pipe is now accessible to the LDV.

Figure 6-1 shows that while the mode shapes approximate those expected, at the mouthpiece end there is an increase in amplitude prior to reaching the clamp. One might have expected that the fixed end of the pipe should be a velocity node due to the clamp restricting any vibration. However, it is clear that in practice the clamps do not provide a totally rigid restriction, instead allowing a degree of ‘play’ in the pipe.

6.3 Decoupling the Lips from the Instrument's Wall

6.3.1 Experimental Method

The section of brass pipe was set-up as in the previous section, with the two ends clamped around their entire circumference. However, in this case, the clamp at the mouthpiece end of the pipe was moved to a position approximately 15 mm in from the pipe end.

The artificial mouth was coupled to the pipe, the air supply activated and the mouth then adjusted until the instrument produced a stable note. The peak velocity amplitude variation along the pipe was measured at twenty-three points at 3 cm intervals and recorded in mms^{-1}.

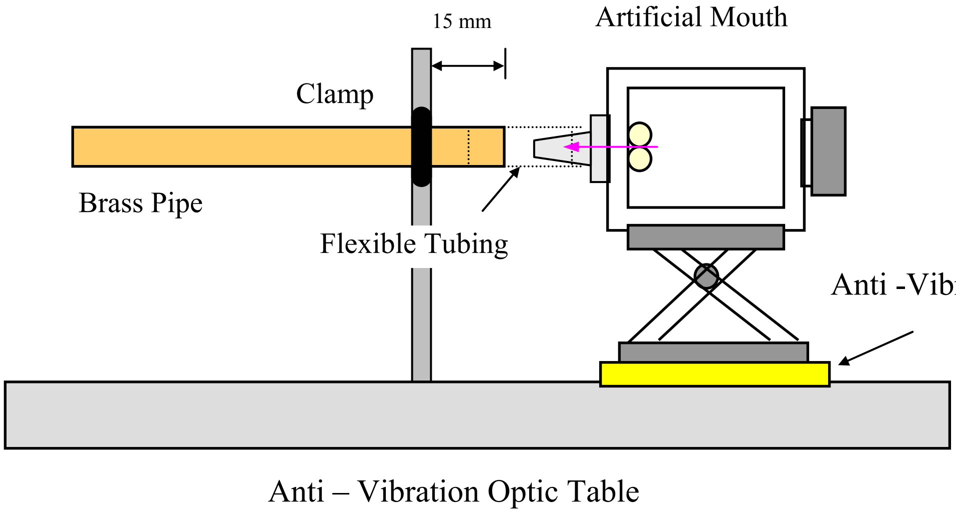
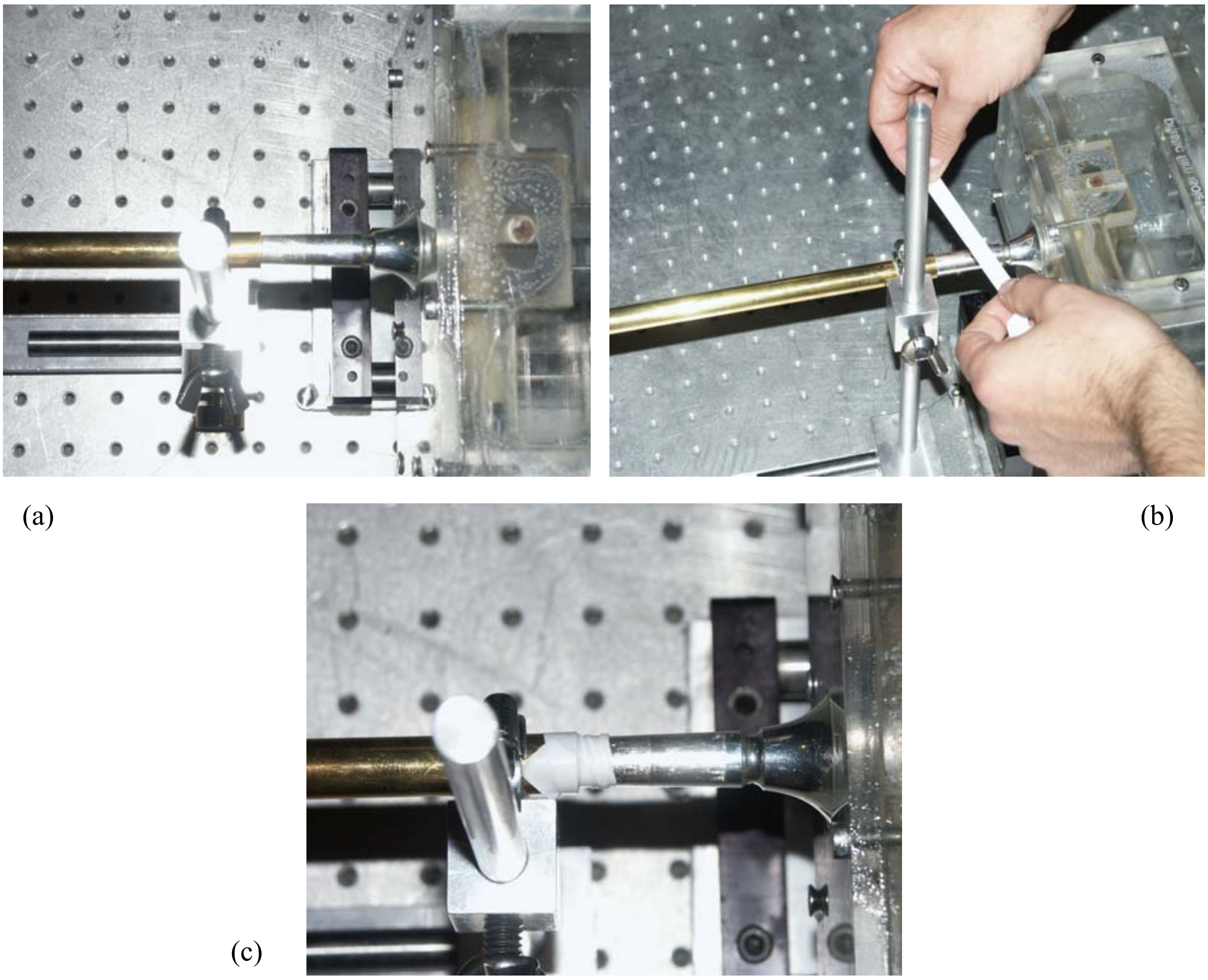


Figure 6-3. Diagram of decoupled pipe with inserted tubing (not to scale)

Without turning off the air supply and with the lips still buzzing, the pipe was decoupled from the mouthpiece. This was accomplished by sliding the artificial mouth backwards away from the clamped pipe section until the end of the mouthpiece was level with the end of the pipe. It was ensured that the finishing position of the mouthpiece held the same relative attitude to the brass pipe as it did while they were coupled and that there was no contact between them. The mouthpiece and pipe were then reconnected by winding PTFE tape (Polytetrafluorethylene tape - a plastic tape used primarily as a thread seal) over the 15 mm exposed section of pipe and the mouthpiece backbore, creating a short flexible tube between them (Figure 6-4). It was anticipated that this would reduce any vibration transmitted from the lips to the pipe via the mouthpiece without significantly altering the strengths and frequencies of the air column resonances.

The velocity amplitude variation along the pipe was derived from LDV measurements as before, with the measurements made at the same locations.



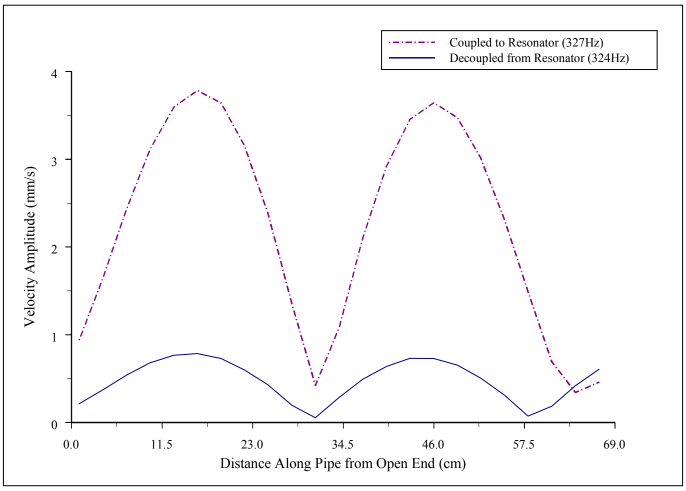
(a) (b) (c)

Figure 6-4. A photographic series, (a) – (c) demonstrating the decoupling of the pipe from the mouthpiece

As the artificial mouth and simple instrument were both situated on the same anti-vibration optical table, an anti-vibration mat was placed between the artificial mouth and the table to inhibit the possible transmission of any vibration to the pipe through the table.

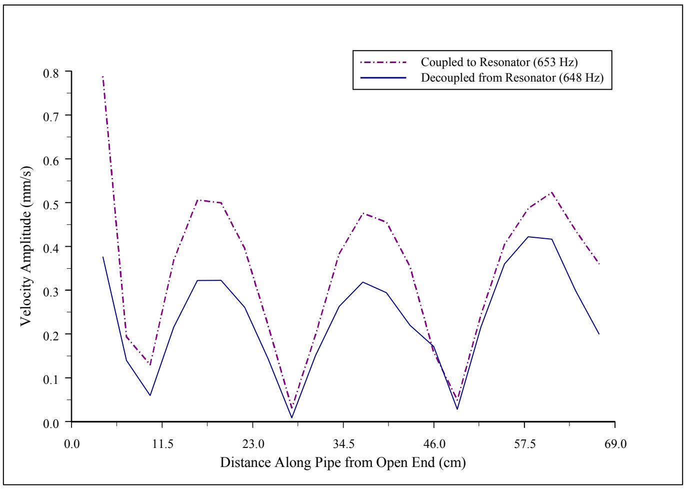
6.3.2 *Experimental Results*

Figure 6-5 shows the velocity amplitude variations along the artificially blown pipe, at (a) 327Hz, (b) 653 Hz and (c) 975 Hz, with the brass pipe regularly coupled to the mouthpiece and then decoupled by the flexible tubing. The plots show a significant reduction in the peak induced velocity amplitudes after insertion of the flexible tube compared with those measured when the brass pipe is normally coupled to the mouthpiece. The effect is most dramatic at the fundamental frequency of 327 Hz (Figure 6-5(a)) with the velocity amplitude peaks reduced to 21% of the peak wall velocities found under normal (coupled) playing conditions. This indicates that, at this frequency especially, the motion of the lips against the mouthpiece is the dominant excitation mechanism by which wall vibrations are excited.

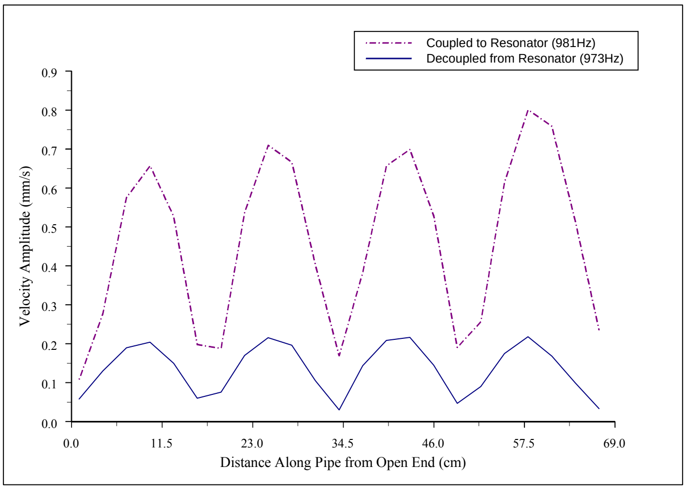


(a)

Figure 6-5. Velocity amplitude variation of pipe when artificially blown - with and without flexible tubing present. First harmonic (a) 327 Hz



(b)



(c)

Figure 6-5. (cont.) Velocity amplitude variation of pipe when artificially blown - with and without flexible tubing present. Second harmonic (b) 653 Hz and third harmonic (c) 975 Hz

Note the slight lowering of the fundamental frequency from 327 Hz to 324 Hz when the flexible tubing is inserted. This is due to the effective length of the instrument being marginally increased

6.4 Decoupling the Air Column from the Instrument's Wall

6.4.1 Experimental Method

The section of brass pipe was set-up under the same conditions stated in Section 6.3.1. The artificial mouth was coupled to the pipe, the air supply activated and the mouth adjusted until the instrument produced a stable note. The peak velocity amplitude variation along the pipe was measured with the LDV at sixty six points at one cm intervals and recorded in mm s^{-1}.

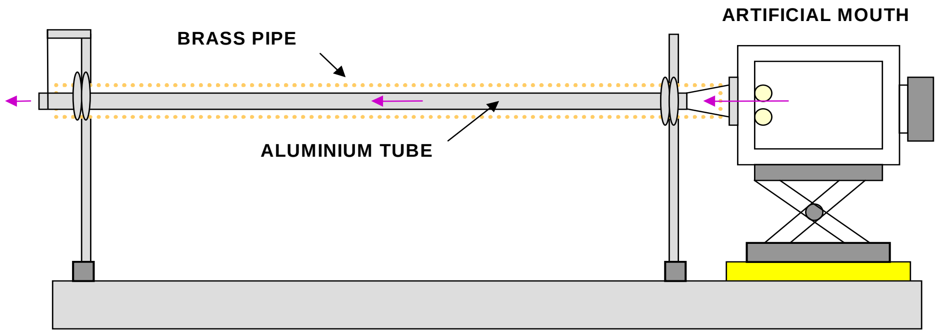
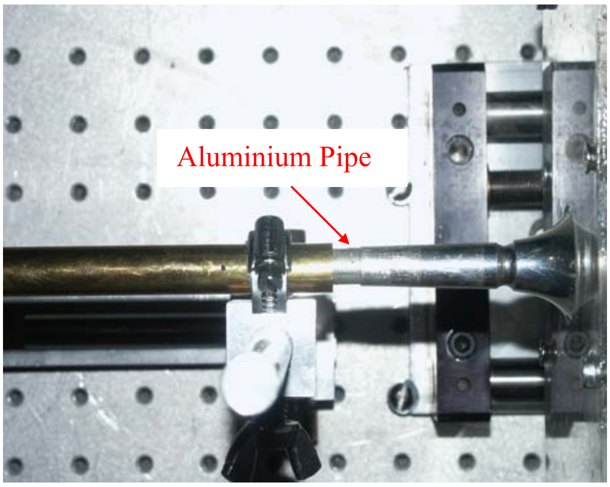


Figure 6-6. Diagram of brass pipe with secondary resonator (aluminium pipe) inserted

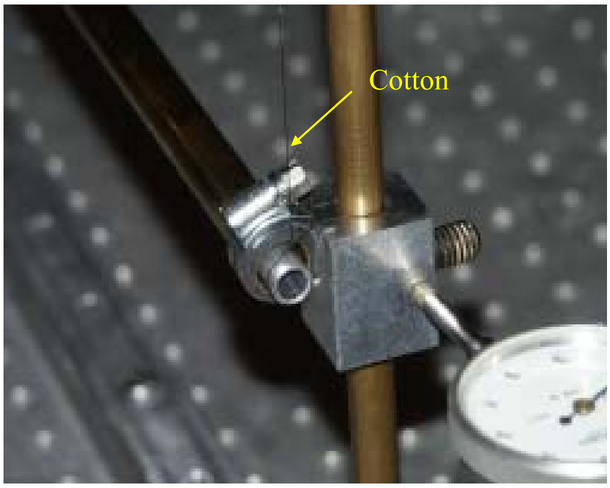
Without turning off the air supply and with the lips still buzzing, the pipe was decoupled from the mouthpiece and an aluminium tube of slightly smaller outside diameter (length 70 cm outside diameter 9.5 mm) was inserted within the brass pipe. The aluminium pipe was fixed to the end of the mouthpiece backbore using a small strip of double-sided adhesive tape and the mouthpiece reinserted into the brass pipe. (The applied force of the mouthpiece into the brass pipe needed to be comparable to the previous coupled LDV measurement. To achieve this, the artificial mouth was mounted on a track system which is described in more detail in Chapter7). The aluminium pipe then became the acoustic resonator of the instrument ensuring that pressure changes within the air column were prevented from acting on the outer pipe and the only excitation experienced by the brass pipe was the motion of the lips against the mouthpiece.

The fixing of the aluminium pipe to the mouthpiece caused its open end to extend beyond the open end of the brass pipe allowing it to be loosely suspended by a loop of cotton (Figure 6-7).

An electrical continuity tester was used to make sure there was no contact between the two pipes (the adhesive tape prevented any contact between the aluminium pipe, mouthpiece and brass pipe).



(a)



(b)

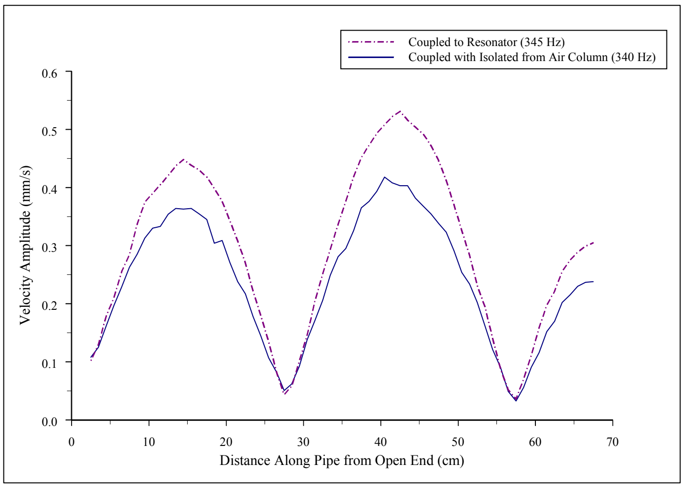
Figure 6-7. Aluminium tube inserted into the instrument. (a) Fitted into mouthpiece (b) Open end suspended by cotton

The velocity amplitude variation along the pipe was derived from LDV measurements as before, repeated at the same locations.

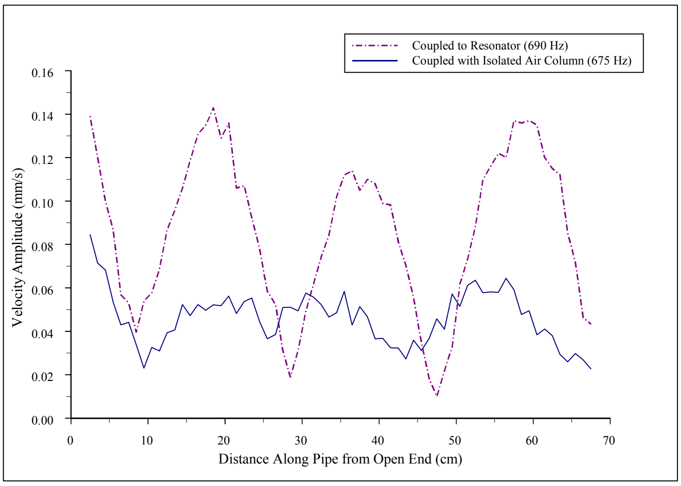
6.4.2 *Experimental Results*

Figure 6-8 compares the velocity amplitude variation along the artificially blown pipe, at the first two harmonics ((a) 340 Hz and (b) 675 Hz), when the brass pipe was regularly coupled to the mouthpiece with the variation when it was coupled to the mouthpiece but isolated from the air column. The third harmonic has not been presented as, at this frequency, noise in the signal prevented the velocity amplitude readings from being adequately discerned.

At the fundamental frequency of 340 Hz (Figure 6.8(a)), the plots indicate that the peak velocity amplitude induced in the wall of the brass pipe when the aluminium pipe was present was 79% of the value measured prior to the insertion of the aluminium pipe. Clearly, the vibrations induced in the wall of the brass pipe when the aluminium pipe was present are still very significant, despite there being no interaction with the air column. So, this again provides a strong indication that the interaction of the lips with the mouthpiece is the dominant excitation mechanism. However, it is also clear that inserting the aluminium pipe has caused some reduction of the velocity amplitudes induced. This could simply be a result of the added mass of the internal alumiunium pipe damping the vibration of the mouthpiece, or it could be that the insertion of the narrower diameter tube with its higher impedance has altered the standing wave feedback on the lips and, consequently, adjusted the manner in which they vibrate. Alternatively, it could imply that the pressure changes within the air column do play a small part in the excitation of the wall vibrations.



(a)



(b)

Figure 6-8. Velocity amplitude variation of pipe when artificially blown - regularly coupled and coupled with isolated air column. Measured harmonics (a) 340 Hz and (b) 675 Hz

At the second harmonic (Figure 6-8(b)), the plots indicate that isolating the air column causes the peak velocity amplitude induced in the walls of the brass pipe to be reduced to 43% of its original value. One explanation for this much larger reduction might be that the frequency of the second harmonic (675 Hz) is close to the mouthpiece resonance frequency (approximately 500 Hz). As was explained in Chapter 2, around the mouthpiece resonance frequency, the impedance peaks of the instrument are strongly modified. Therefore, any effect that the pressure changes within the air column might have on the walls of the instrument would be expected to be greatest at such frequencies. However, the plotted mouthpiece resonance (Figure 5-5) appears to peak at a frequency approximately equidistant between the first and second harmonics yet the first harmonic seems to be unaffected by this modification.

Note that there is once again a lowering of the fundamental frequency from 345 Hz to 340 Hz when the aluminium pipe is inserted. This could also be due to the effective length of the instrument being marginally increased as the aluminium pipe is longer than the brass pipe and is fixed to the very end of the mouthpiece backbore.

6.5 Lip Stability Concerns over Repeated Pipe Decoupling

A problem encountered in the execution of this series of experiments was that of maintaining the lip vibration when decoupling the air column from, and then reattaching it to, the mouthpiece. Detaching the acoustic resonator causes the feedback of the standing waves experienced at the lips to be removed. This could have consequences in later comparison measurements between instruments of different materials where the pipe would be removed repeatedly while sounding the same note.

In an attempt to manage the problem, prior to the decoupling of the pipe, the fundamental frequency and amplitude of the note produced was monitored. Then, after the pipe had been decoupled and reattached, it could be checked whether the fundamental frequency and amplitude had been maintained. If the parameters had changed, the experiments were repeated until the situation was reached where they were maintained.

6.6 Conclusions

Experiments have shown that when a simple wind instrument, consisting of a mouthpiece and section of brass piping, is artificially blown mechanical wall resonances are excited. The results presented suggest that, though the air column resonances may have a slight effect, it is the motion of the lips against the mouthpiece that is the main cause of the wall vibrations when the instrument is blown.

It is possible a combination of the two processes, with the motion of the lips against the mouthpiece dominant, produces the maximum vibrational amplitude. This is a matter for further experimentation.

If the wall vibrations of a brass instrument do have an effect on the tone produced, the amplitude of these vibrations is likely to be very important. Different musicians use differing amounts of pressure of the lips in the mouthpiece to produce the same note. Consequently, different musicians will induce different amplitudes of wall vibration. Another factor which will affect the amplitude of the wall vibrations is the material from which the instrument is manufactured. The effect of the wall material and the wall thickness on the amplitude of the vibrations induced when a lip-reed instrument is blown is investigated in the next Chapter.

Chapter 7

Comparing Wall Material and Wall Thickness

7.1 Introduction

In this chapter a series of comparative experiments to investigate the effect of the material of construction and wall thickness on the vibrations induced in the walls of lip-reed instruments are reported. Structural response measurements, carried out on a number of simple instruments (each of which comprises a trombone mouthpiece coupled to a pipe of different wall material or thickness) are presented and discussed. Measurements of the vibrations induced in the walls of these instruments when they are artificially blown are then described and compared.

7.2 Instruments used in the Study

The simple instruments employed in the comparative experiments formed two groups. The first group had practically identical geometry but were made from different materials. The second group were all constructed from the same brass alloy and had the same internal dimensions but their wall thicknesses were different. All of the pipes utilised in the study were fashioned by the process of drawing. Seamless pipes were required to negate any possible influence of the played note through variations in bore roughness.

7.2.1 Differing Wall Material

Cylindrical pipes made from five different materials - brass, copper, aluminium alloy, stainless steel and titanium alloy - were acquired to test the vibrational influence of wall material. The pipes were all commercially available (although not from a single manufacturer) and each had an external diameter of 12.7 mm and a wall thickness of 0.71 mm. This internal diameter was chosen as it would reasonably accommodate a trombone mouthpiece and the wall thickness was chosen as it approximately matched that used in regular instrument manufacture. Purchased sections were accurately cut down to one metre lengths and checked to make sure that they were straight and had no obvious indentations or defects that could adversely affect the LDV measurements. The interior of each was then cleaned and end burrs removed. The five pipes can be seen in Figure 7-1.

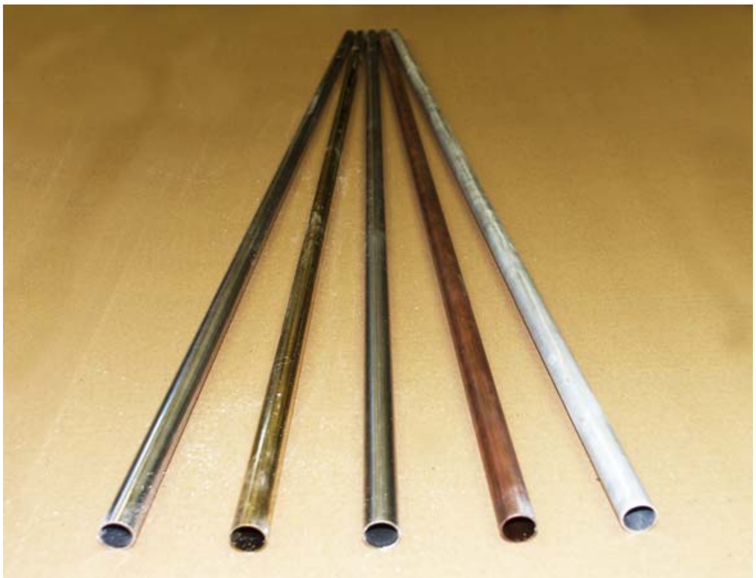


Figure 7-1. The five pipes of varying wall material used in the study

7.2.2 Differing Wall Thickness

To test the vibrational influence of the wall thickness, it was originally intended to use a single pipe and to use a lathe to remove material from the wall between measurements. However, this was deemed impossible, as a thin walled pipe would bow in the clamps of the lathe causing an uneven thickness to be produced along the pipe's length. Therefore, lengths drawn to predetermined thicknesses were used instead.

A number of pipes were custom-made by Michael Rath Brass Instruments, a company specialising in the handcrafting and repair of trombones. The pipes were fashioned from the same brass alloy with identical lengths (78 cm) and almost identical internal bore diameters (approximately 12.63 mm). A variation of a hundredth of a millimetre was actually found between the bore diameters, though this may be due to some wear at the end of the pipe or a slight error in the calibration of the digital calipers.

Figure 7-2. Pipes showing varying wall thickness and mouthpiece/ mouthpiece adaptor

Each pipe was fitted with a threaded collar onto which an adaptor, designed to fit the trombone mouthpiece, could be screwed. This allowed the same connection to be made between each test pipe and the mouthpiece helping ensure a good, repeatable connection.

As the adaptor extended for approximately 14.5 cm within the pipe, measurements along the wall of each pipe would extend over the length of the adaptor.

Three of these pipes – with wall thicknesses of 0.335 mm, 0.535 mm and 0.630 mm – were used in the experiments. These wall thicknesses are comparable to those found in manufactured instruments.

7.3 Adapting the Experimental Set-up for Comparative Measurements

To investigate variations in the wall vibrations in instruments made from different materials and with different wall thicknesses, it was intended to carry out structural response measurements, and measurements of the wall vibrations induced when the instruments are artificially blown, using the basic techniques outlined in the previous chapters. However, as the instruments had to be repeatedly changed during the course of both sets of experiments, modifications to the experimental set-up were necessary. These were to ensure that all the instruments were measured under the exact same experimental conditions. In other words, to ensure that the excitation force was always applied at the same location with the same force, and that the clamping conditions were identical.

7.3.1 Adaptation for Consistent Excitation Force

For the structural response measurements, to ensure that the force exerted by the mechanical shaker onto each pipe was the same for each instrument, the shaker was permanently fixed to a potentiometer system (Figure 7-3), which was itself attached to a metal support framework. This potentiometer allowed the vertical movement of the shaker to be measured in terms of resistance. Readings were taken using a Thurlby digital multimeter 1503-HA with a resolution of one milliohm which equated to a vertical movement by the shaker of approximately 0.01 mm. The potentiometer system provided a means of ensuring that the loading of the shaker on to each pipe remained the same for each instrument.

The supporting framework negated lateral and longitudinal movement of the shaker which, along with the restricting clamp at the open end of the pipe, allowed the needle attachment to be positioned at the same point of the instrument with a high degree of accuracy.

Figure 7-3. Photograph showing the mechanical shaker/potentiometer system

For the artificially blown measurements, to ensure that the force exerted by the vibrating lips (via the mouthpiece) on the pipe was the same for each instrument, the artificial mouth was firmly attached to a moveable track with a ‘fine’ horizontal adjustment allowing the mouthpiece insertion to be repeatable in terms of angle and pressure. The insertion pressure was monitored at the open end of the pipe by connecting a Mercer Type 16 dial gauge (graduated with one thousandth of an inch increments) to the clamp holding the pipe.

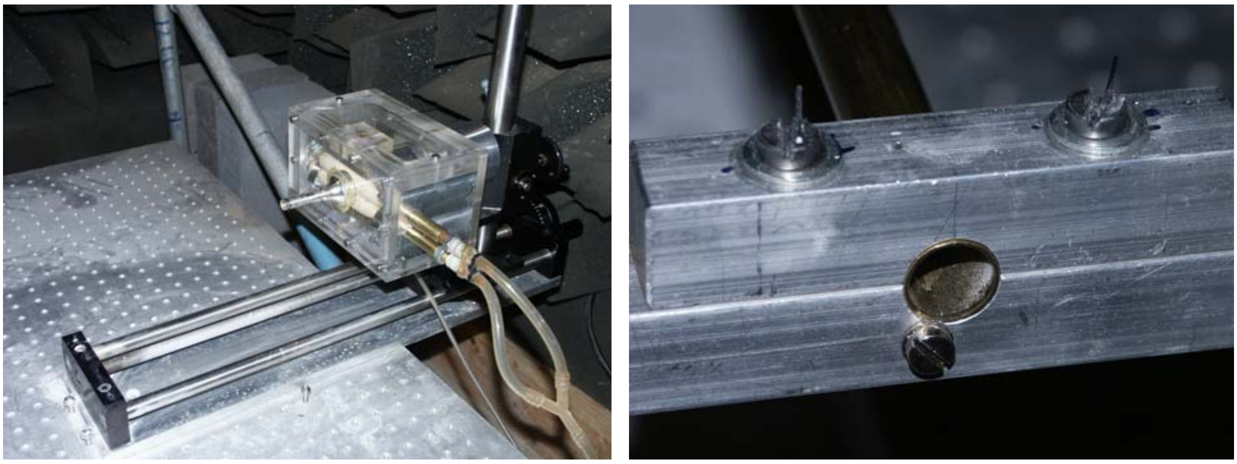


Figure 7-4. Photographs of track and clamp used to secure pipes for measurement

7.3.2 Alterations to Clamping Conditions

For both the structural response and the artificially blown measurements, a new clamping arrangement was implemented. The clamp was removed from the mouthpiece end of the pipe, leaving the inserted mouthpiece to hold the pipe rigidly in position (the dial gauge system described in the previous section ensured that each pipe was held under identical conditions). The removal of the clamp prevented the problem of having to loosen and then tighten the clamp by the same degree between measurements on different instruments.

A clamp was retained at the open end of the pipe. For the measurements on the pipes of different wall materials, a clamp designed to hold pipes of outside diameter 12.7 mm was used. The clamp has a small lip which overhangs the edge of the hole through which the pipe is inserted. This lip prevents the pipe from being pushed through the aperture of the restraining clamp (Figure 7-4). A similar clamp was used for the measurements on the pipes of differing wall thickness. However, the clamp for this experiment required the pipes of varying outside diameter to be held to the same degree. This was achieved by constructing a split ring metal grommet for each pipe. The grommets were manufactured to have the same outside diameter, enabling a single clamp to be produced which was suitable for all the pipes.

7.4 Determination of Structural Modes

7.4.1 Experimental Method/Procedure

The experimental method used to determine the structural responses of the simple instruments (each instrument comprising a trombone mouthpiece coupled to a pipe of different wall material or thickness) was virtually the same as that described in Chapter 5.

For each instrument under investigation, the section of pipe was fixed horizontally on an anti-vibration optic table housed in an anechoic chamber. It was held rigidly at one end by the trombone mouthpiece (embedded in the front plate of the artificial mouth) and at the other end by a fixed clamp. To ensure that the force exerted on the pipe by the mouthpiece was the same for all the instruments, the position of the artificial mouth was adjusted until the dial gauge (described in Section 7.3.1) advanced through five increments. This force ensured a firm contact without damaging the end of the pipe or mouthpiece.

In order to excite the mechanical resonances, the instruments of differing wall material were driven with a Chirp signal using the Ling Dynamics shaker with needle attachment. The instruments of varying wall thickness were driven at individual frequencies from 5 Hz to 1.4 kHz (at 5 Hz intervals) using the same Ling Dynamics shaker. The needle was positioned 2 cm from the mouthpiece. To locate the needle, the shaker was lowered until the needle made contact with the pipe (the first point of contact was verified using a continuity tester). The shaker was then lowered through a further 0.4 mm (measured using the potentiometer described in Section 7.3.1). This ensured that the shaker remained in contact with the pipe over all the frequencies to be excited.

To determine the velocity amplitudes excited in the pipe walls at each of the frequencies driven by the shaker, the Polytec LDV system was utilized. It was mounted vertically above the simple instrument on a purpose built track system. This track system allowed the sensor head to traverse the length of the instrument and, in this way, velocity amplitude measurements were made at 3 cm intervals along the pipe at each of the frequencies excited by the shaker.

The frequency sweep signal sent to the shaker was provided by a Matlab program written within the department. The same program also recorded the velocity amplitudes output by the vibrometer.

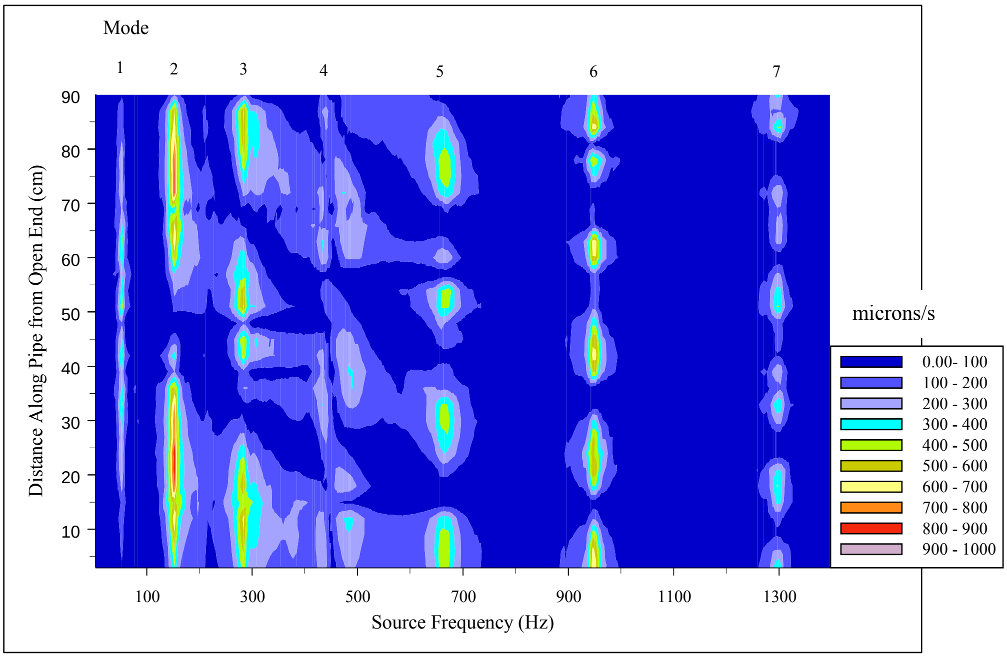
Structural response measurements were carried out in this manner for the five simple instruments with different wall materials and for the three simple instruments with different wall thicknesses.

It should be noted that, due to the positioning of the mechanical shaker over the end of the pipe the end velocity amplitude peak for the higher resonance modes are obscured and so absent from the 2D plots.

7.4.2 Results

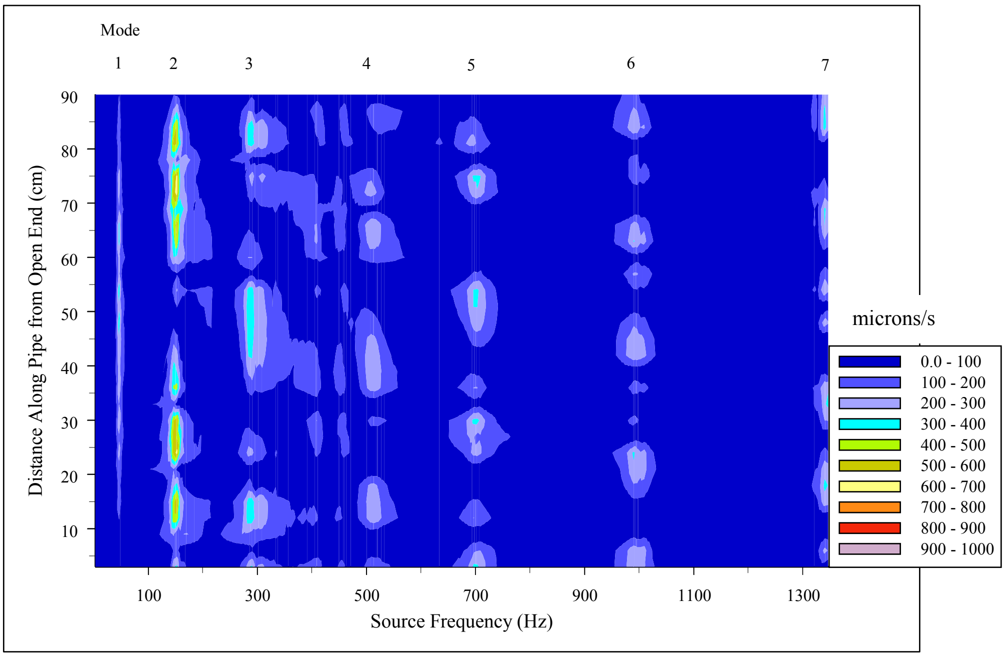
7.4.2.1 Wall Material

Titanium Alloy



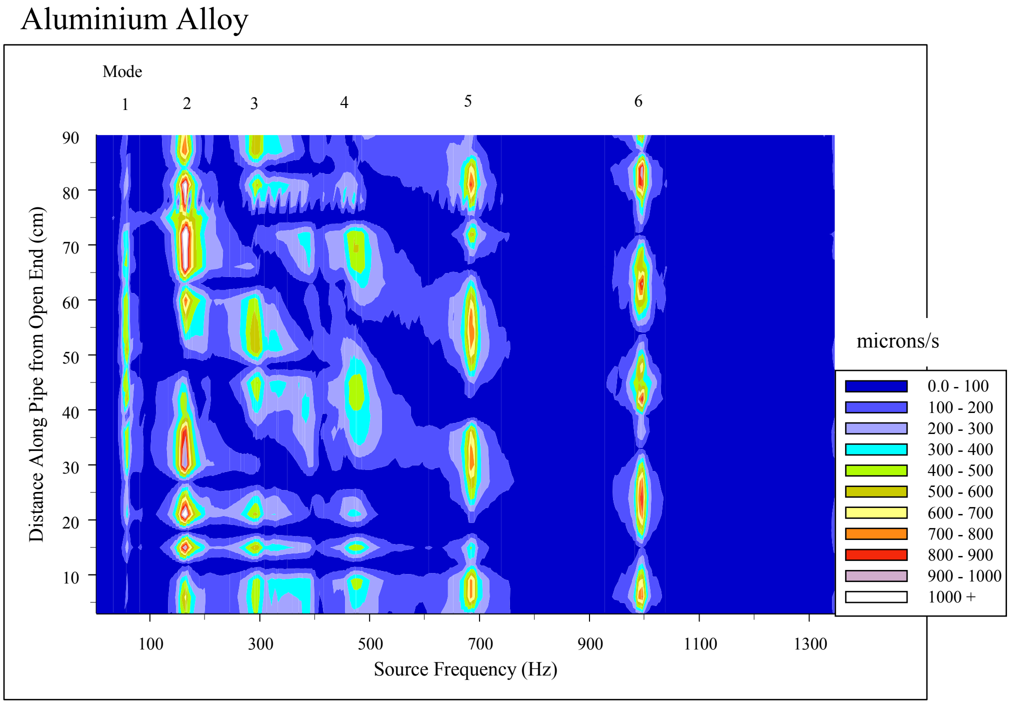
(a)

Steel

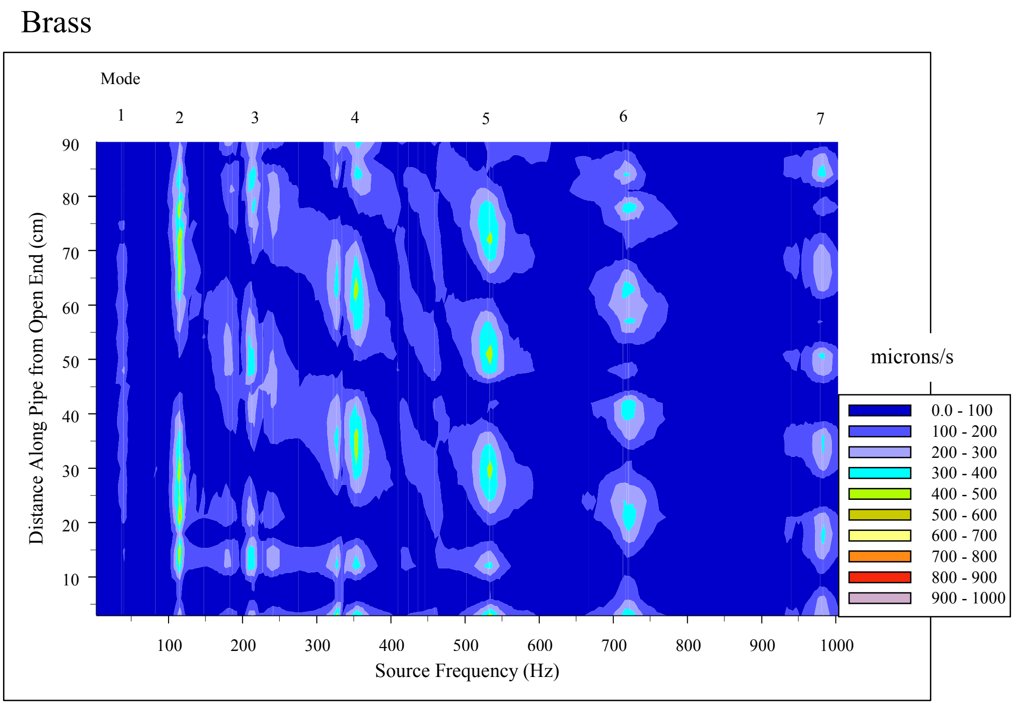


(b)

Figure 7-5. 2D contour map of (a) titanium alloy and (b) steel

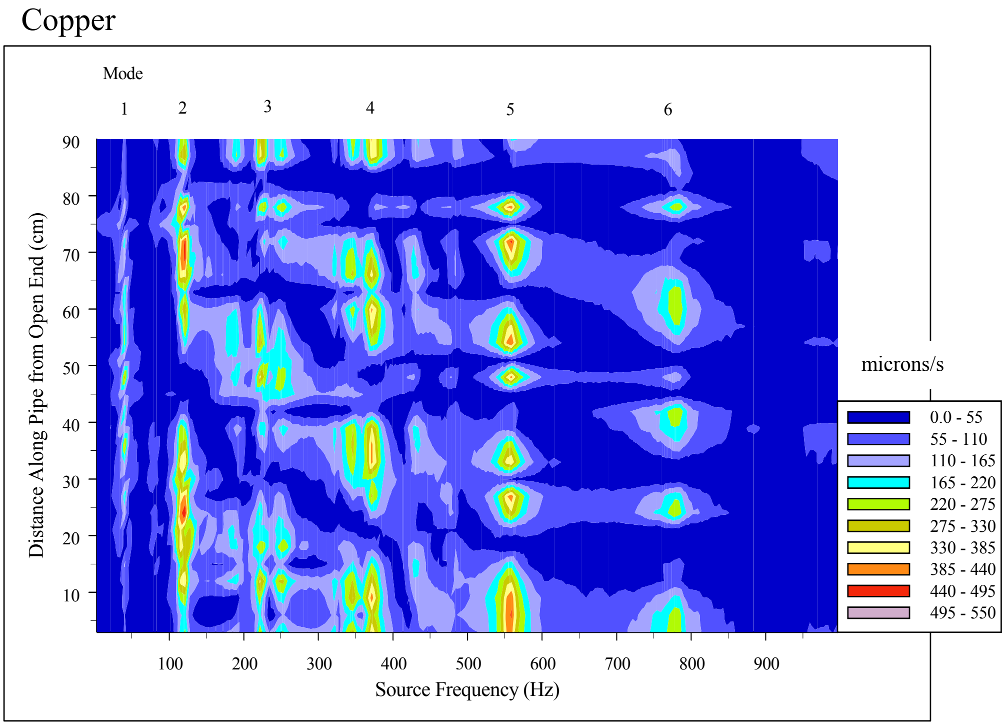


(c)



(d)

Figure 7-5(cont.). 2D contour map of (c) aluminium alloy and (d) brass



(e)

Figure 7-5(cont.). 2D contour map of (e) copper. (Change in legend compared to previous)

The structural vibration responses for the five different wall materials can be seen in the 2D contour plots, Figure 7-5(a) - (e). From these graphs, the first seven resonance frequencies for each of the five different wall materials have been extracted. These are recorded in Table 7-1.

(Hz)	Mode 1	Mode 2	Mode 3	Mode 4	Mode 5	Mode 6	Mode 7
Titanium	53	153	284	484	668	946	1297
Steel	47	153	290	512	700	991	1340
Aluminium	59	165	292	472	684	997	1359
Brass	40	116	209	353	534	720	984
Copper	38	121	222	371	559	778	1043

Table 7-1. Table of resonance frequencies for the first seven modes of vibration for each of the pipes of different wall material

This dependence of the wall vibrations on the material of construction can be seen quite clearly either through examination of Figure 7-5(a) – (e) or by inspection of Table 7-1. The difference in frequency between equivalent modes for each material generally increases with frequency, with a maximum difference of 375 Hz (between brass and aluminium) occurring at the seventh mode.

Variations in the amplitude of vibration can also be made out between the different wall materials. This is to be expected, as the stiffness will vary since the Young's Modulus is different for each material. The maximum velocity amplitudes occur in the structural response measurement made on the aluminium instrument, the lowest in measurements made on the brass and copper.

As the modes of vibration for each material occur at different frequencies the response of the mechanical shaker has to be taken into consideration. Table 7-2 shows the maximum amplitudes of the first seven modes for each material after dividing out the shaker response at each frequency. The response of the mechanical shaker over the frequency range of interest can be seen in Figure 7-6.

microns/s	Mode 1	Mode 2	Mode 3	Mode 4	Mode 5	Mode 6	Mode 7
Titanium	17.6	38.2	80.9	126.8	388.9	1306.8	1516.7
Steel	12.7	28.2	52.6	130.6	299.2	666.0	1237.9
Aluminium	20.0	60.7	81.9	189.2	710.2	2084.6	1874.5
Brass	7.8	20.2	26.1	91.6	215.5	378.9	708.3
Copper	10.5	18.3	27.3	92.9	250.2	322.1	1010.4

Table 7-2. Table of maximum amplitudes for the first seven modes of vibration for each of the pipes of different wall material.

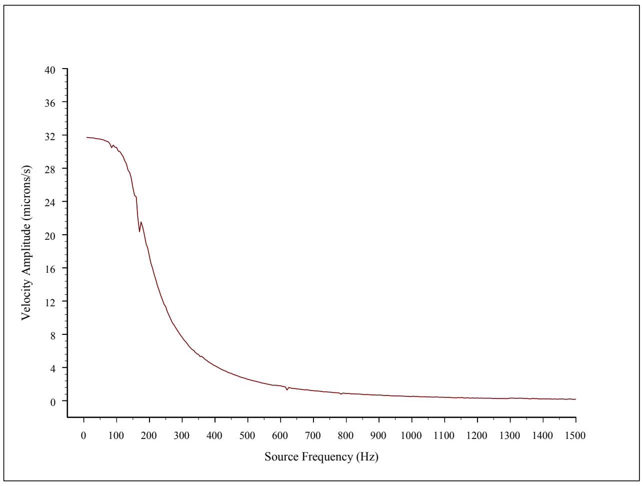
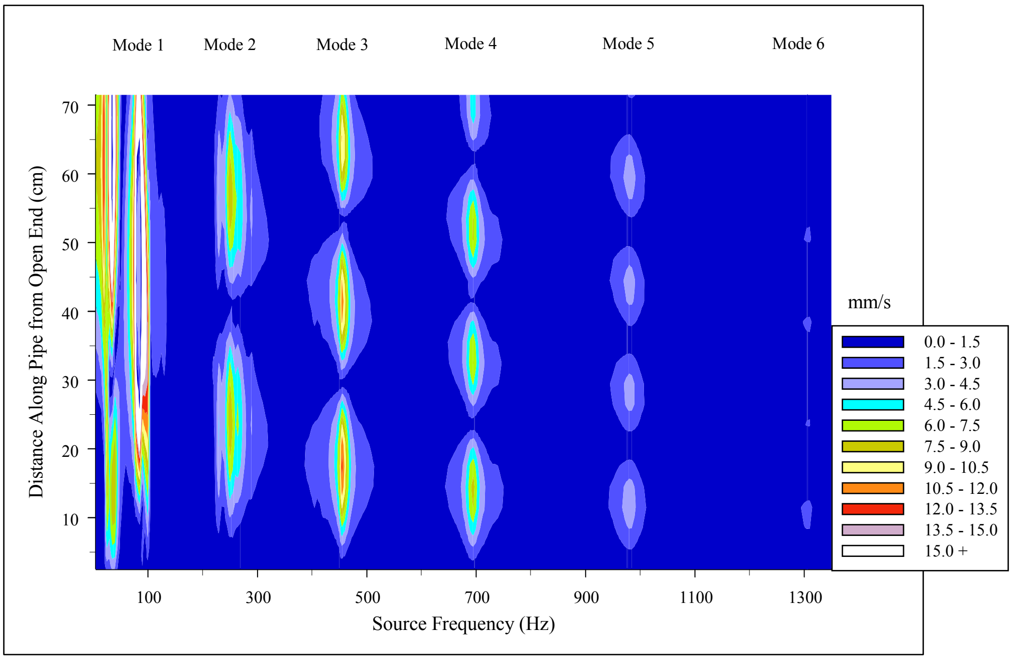
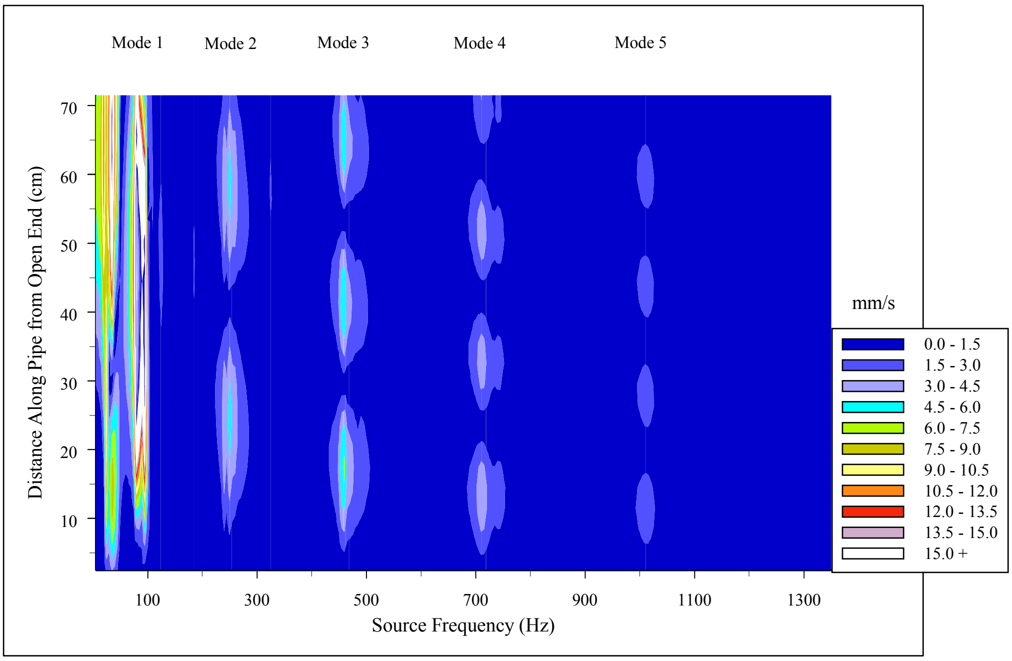


Figure 7-6. Shaker response over the frequency range of interest.

7.4.2.2 Wall Thickness

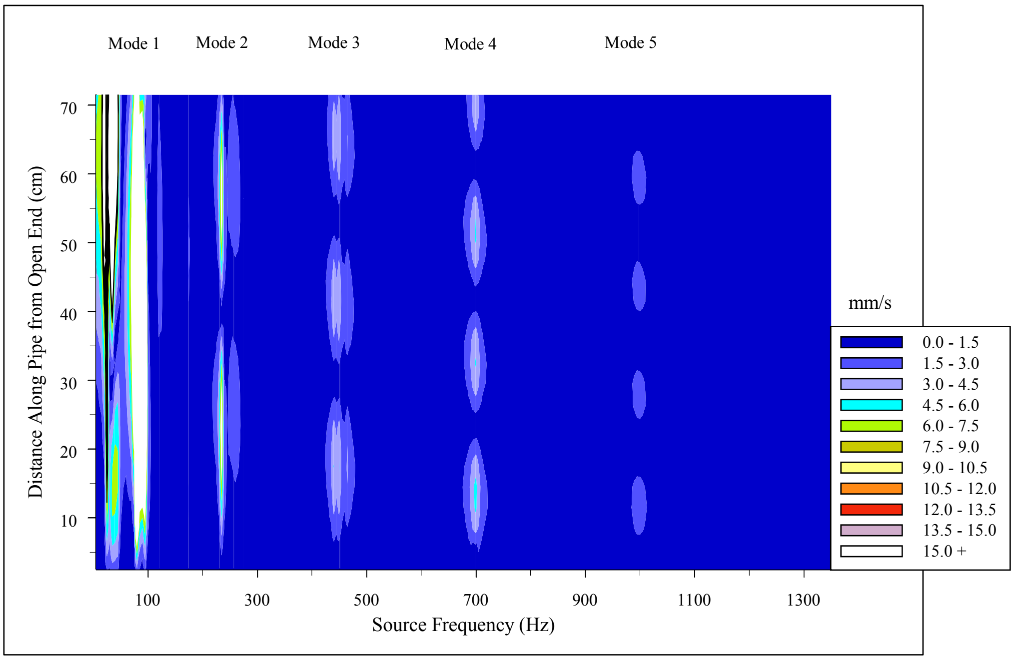


(a)



(b)

Figure 7-7. 2D contour map of (a) thin and (b) medium thickness brass.



(C)

Figure 7-7(cont.). 2D contour map of (c) thick brass

The structural vibration responses for the three brass pipe wall thicknesses can be seen in the 2D contour plots shown in Figure 7-7(a) - (c). Again, from these graphs, the first six resonance frequencies for each of the three different wall thicknesses have been extracted and are recorded in Table 7-3 for ease of comparison.

Brass	Mode 1	Mode 2	Mode 3	Mode 4	Mode 5	Mode 6
Thin	85	250	455	695	980	1305
Medium	80	250	460	710	1010	1355
Thick	80	235	450	700	1000	1330

Table 7-3. *Table of resonance frequencies for the first six modes of vibration for each of the pipes of different wall thickness*

The plots for wall thickness all show very well defined modes of vibration where the resonance frequency and mode shape can be easily determined. Differences in the velocity amplitudes at the resonance peaks can be clearly made out with, as might be expected, the thinnest pipe having the greatest velocity amplitudes and the thickest pipe having the smallest.

Variations in the frequencies at which the resonance peaks occur are harder to discern from the plots. Such differences can, however, be established from Table 7-3 which gives the resonance frequencies for all three brass pipe wall thicknesses measured. Examination of the table reveals there is some shifting of the resonance frequencies which becomes more pronounced at higher frequencies. However, this change in resonance frequencies doesn't have such a straightforward dependency on the wall thickness. In general, the frequency at which a given bending mode occurs is highest for the medium thickness pipe. This is certainly true from the third bending mode upwards. For the lowest two bending modes, the thinnest pipe has the higher resonance frequencies. This result was confirmed by a repetition of the experiment, which gave a good agreement with the previous results.

7.5 Determination of Induced Wall Vibration through Artificial Blowing

7.5.1 Experimental Method/Procedure

Again, the experimental procedure used to measure the vibrations induced in the walls of the instruments of different material and wall thickness was virtually the same as that described in Chapter 5.

For each instrument under investigation, the pipe was clamped to the optic table and coupled to the artificial mouth in the same way as described in section 7.4.1. However, in this case, the shaker was removed from the set-up. The air-supply to the artificial mouth was then activated and the lips adjusted until a discernible stable note was heard. With the laser Doppler vibrometer positioned in the centre of its track, its scanning capabilities were employed and a single sweep was made to measure the velocity amplitudes at 3 cm intervals along the length of the pipe. During the vibrometer scan, a microphone positioned 50 cm from the open end of the pipe monitored the sound produced by the instrument and a micro-manometer monitored the pressure inside the artificial mouth.

This was repeated for the five simple instruments with different wall materials and for the three simple instruments with different wall thicknesses. When one pipe was removed and replaced with another of different material or wall thickness, the microphone and micro-manometer readings were observed to ensure that both the fundamental frequency of the note produced and the mouth pressure were maintained approximately constant over all the instrument measurements.

7.5.2 Results

7.5.2.1 Wall Material

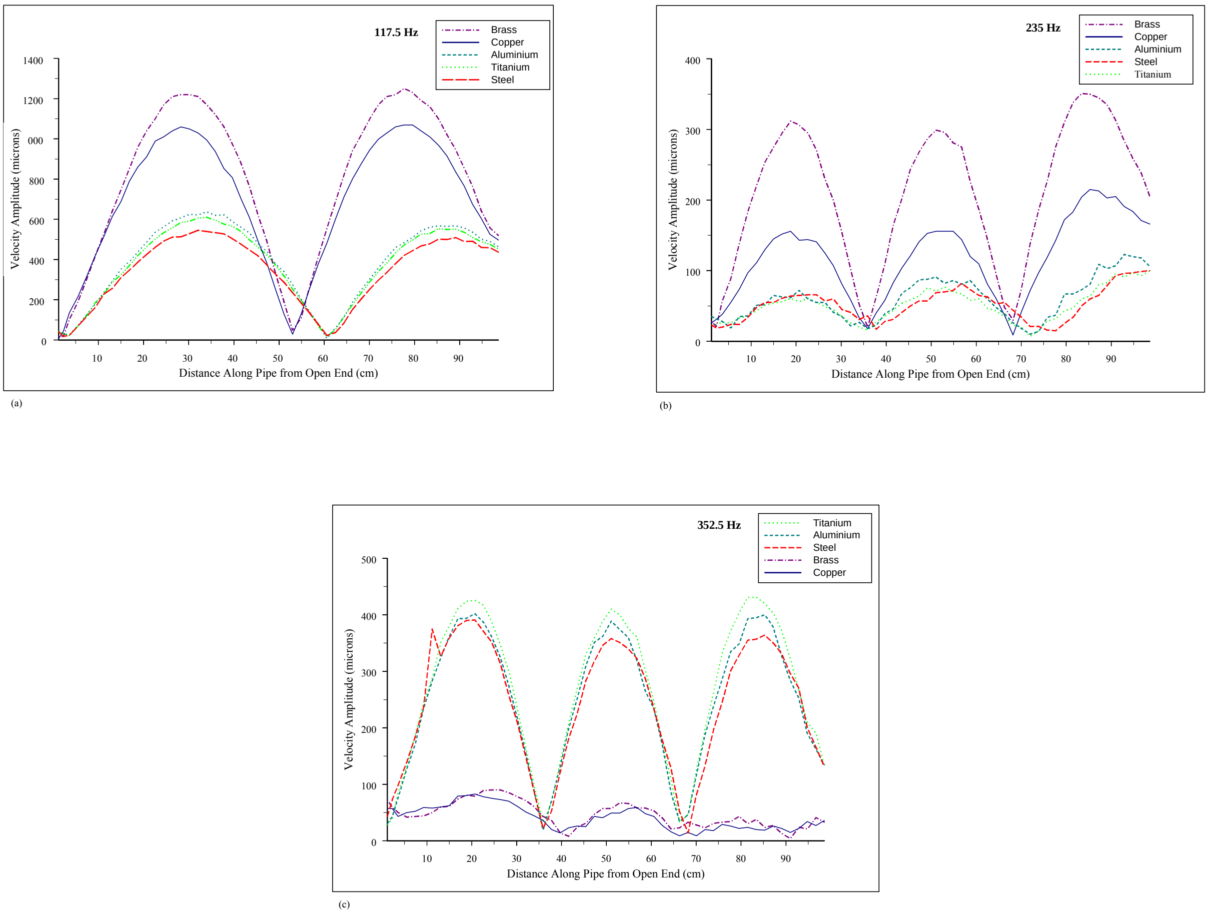


Figure 7-8. Peak velocity amplitude variation along the instrument for each wall material when artificially blown at specific note. Partials (a) 117.5 Hz, (b) 235 Hz, and 352.5 Hz

Figure 7-8(a) – (c) show the velocity amplitudes induced in the walls of the five instruments of different wall material at the partials of the played note. At the fundamental frequency of 117.5 Hz, the brass pipe exhibits the greatest velocity amplitudes with those measured in the copper pipe also significant. The velocity amplitudes induced in the walls of the aluminium, steel and titanium pipes are all of a similar magnitude to each other but are much smaller than those found in the brass and copper pipes. At the second harmonic of 235 Hz, whilst all the velocity amplitudes are much reduced, the same trend is seen, with the wall vibrations induced in the brass and copper pipes again greater than those induced in the other pipes. At the third harmonic of 352.5 Hz, however, this trend is reversed with the brass and copper pipes having negligible wall vibrations while those induced in the steel, aluminium and titanium pipes are stronger and again of a similar intensity to each other.

The most likely reason for these observed differences in the amplitudes of the induced wall vibrations are differences in the structural responses of the five pipes of different materials at the harmonics of the played note. To help investigate this, Table 7-4 shows the maximum velocity amplitudes measured for each of the five pipes at 117.5 Hz, 235 Hz and 352.5 Hz (extracted from the 2D contour plots of Figure 7-5).

	Brass	Copper	Aluminium	Titanium	Steel
117 Hz	598	474	130	109	90
235 Hz	251	239	241	187	85
352 Hz	436	270	350	246	140

Table 7-4. Peak velocity amplitude (microns/s) for the structural resonances taken at the induced harmonic frequencies. Taken for the five different wall materials

At 117.5 Hz, the brass pipe has the strongest structural response closely followed by that of the copper pipe. The structural responses of the other three pipes are all significantly weaker. At this frequency at least, the differences in the wall vibrations induced when the instruments were artificially blown appear to be explained by differences in the structural responses of those instruments.

At 235 Hz, the brass pipe again has the strongest structural response. However, both the copper and aluminum pipes have structural responses which are nearly as large. The response of the titanium pipe is somewhat weaker whilst that the steel pipe has the weakest response at this frequency. So, again, the differences in the wall vibrations induced when the instruments were blown can be broadly explained by the differences in the structural responses. Certainly, the brass and copper pipes exhibited the largest induced wall vibrations and these pipes have amongst the strongest structural responses. The anomaly is the aluminium pipe. Its structural response at 235 Hz is comparable with those of the copper and brass pipes, but the vibrations induced in the walls of the instrument when blown were smaller than those induced in the walls of the copper and brass instruments.

At 352.5 Hz, the relationship between the amplitude of the wall vibrations induced when the instruments were blown and the strengths of the structural responses of the pipes of different material is less obvious. The brass pipe has the largest structural response yet the wall vibrations measured when the brass instrument was blown were amongst the smallest. Conversely, the steel pipe has the weakest structural response at this frequency, yet the induced wall vibrations when the steel instrument was blown were amongst the largest.

7.5.2.2 Wall Thickness

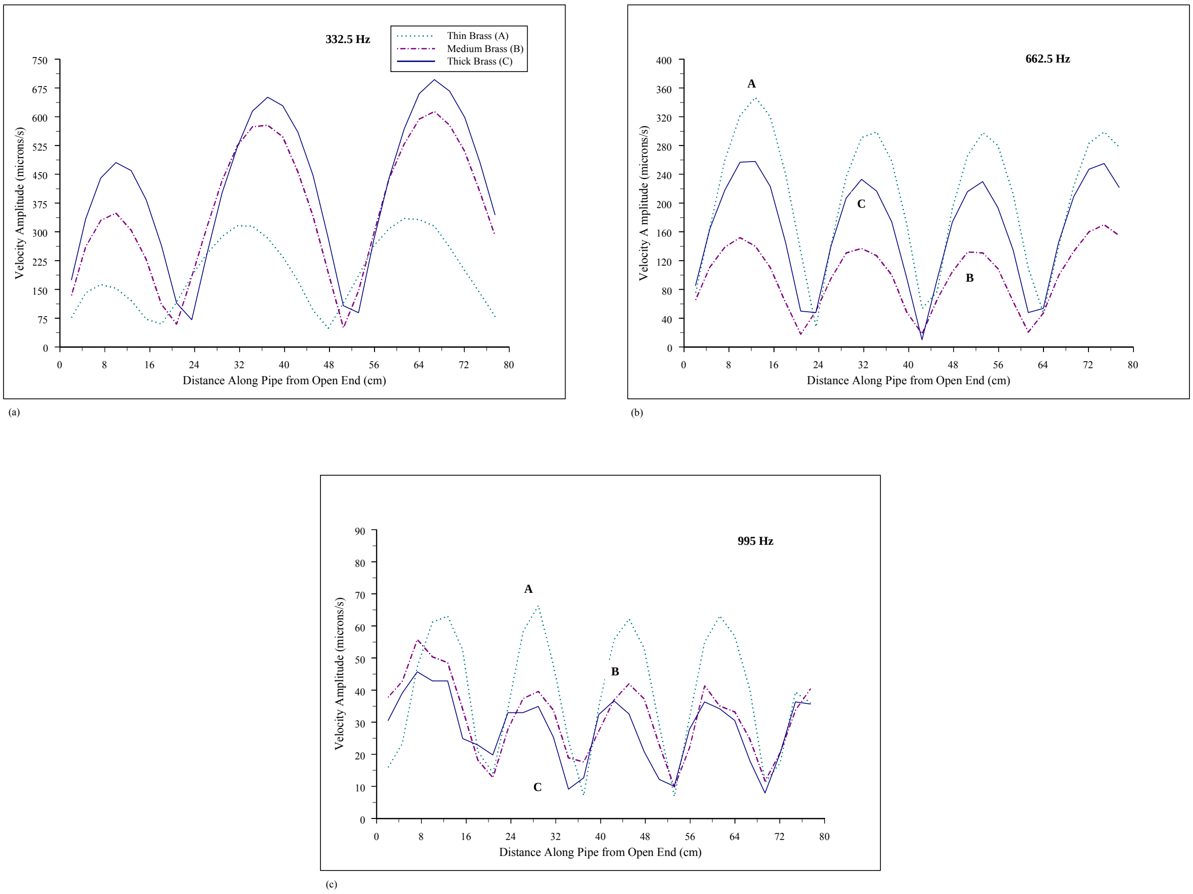


Figure 7-9. Peak velocity amplitude variation along the instrument for each wall thickness when artificially blown at specific note. Partials (a) 332 Hz, (b) 664 Hz, and 996 Hz

Figure 7-9(a) – (c) show the velocity amplitudes induced in the walls of the three brass instruments with different wall thickness at three harmonics of the played note. At the fundamental frequency of 332 Hz, the thickest pipe has the largest velocity amplitudes, reaching a maximum of 700 μm/s approximately 68 cm from the open end. The velocity amplitudes measured in the pipe of medium thickness are only slightly smaller than this whilst those induced in the thinnest pipe are substantially lower, reaching only 340 μm/s (less than half the amplitudes observed in the thickest pipe).

Although the wall vibrations in all three pipes are generally weaker at the second harmonic of 664 Hz, in the thinnest pipe the maximum velocity amplitude observed was approximately 350 μm/s, which is actually greater than the velocity amplitudes induced in this same pipe at the fundamental frequency. Indeed at the second harmonic, it is the thinnest pipe which experiences the strongest wall vibrations. The vibrational velocities induced in the thickest pipe and in the medium thickness pipe are approximately 75 % and 45 % respectively of those observed in the thinnest pipe.

At the third harmonic of 996 Hz, all the induced wall vibrations are very weak. It is again the thinnest pipe which experiences the greatest velocity amplitudes. The velocity amplitudes induced in the two thicker pipes are approximately the same as each other but are only around half the size of those seen in the thinnest pipe.

Just as with the instruments of different wall material, the most likely reason for these observed differences in the amplitudes of the induced wall vibrations are differences in the structural responses of the three pipes of different wall thickness at the harmonics of the played note. To help investigate this, Table 7-5 shows the maximum velocity amplitudes measured for each of the three pipes at 332 Hz, 664 Hz and 996 Hz (extracted from the 2D contour plots of Figure 7-7).

Brass	Thin	Medium	Thick
332 Hz	1.42	1.42	1.40
664 Hz	2.17	1.0	1.04
996 Hz	2.84	1.70	2.25

Table 7-5. Peak velocity amplitude (mm/s) for the structural resonances taken at the induced harmonic frequencies. Taken for the three different wall thicknesses.

Table 7-5 reveals that, at the fundamental frequency of 332 Hz, the structural responses of the three pipes are practically identical. While the wall vibrations induced when the instruments were blown were quite similar in amplitude for the brass pipes with walls of medium and large thicknesses, the vibrations were much smaller for the thin walled pipe. So, at this frequency, the differences in the induced wall vibrations are only partly explained by the structural responses of the pipes.

At the second harmonic of 664 Hz, the structural responses of the thickest two pipes are still practically identical. However, the structural response of the thinnest pipe is now significantly greater. This again partly explains the differences in the wall vibrations induced in the three pipes when the different instruments were artificially blown. The velocity amplitudes measured when the instrument with thin walls was blown were much larger than those recorded for the instruments with thicker walls. However, from the structural response measurements, it might have been expected that when the instruments with medium and thick walls were blown, the induced wall vibrations would have been of similar magnitudes. Instead, the thicker walled instrument vibrated more strongly.

At the third harmonic of 996 Hz, the structural response of the thinnest pipe is again the strongest. The thickest pipe has a somewhat weaker structural response but the pipe of medium thickness has the weakest response at this frequency. Again, this broadly explains the induced wall vibration results where the instrument with the thinnest walls was found to exhibit the strongest wall vibrations when blown.

7.6 Conclusions

The playing consistency offered by the artificial mouth has allowed the wall vibrations excited when simple instruments, manufactured from different materials and with different wall thicknesses, are blown to be measured and compared. The results reveal that the magnitude of the induced wall vibrations clearly depends on both the material from which the instrument is constructed and the thickness of the walls. The differences in the velocity amplitudes measured in the walls of such artificially blown instruments can be broadly explained by differences in the structural responses of the instruments at the frequencies present in the played note. However, at certain frequencies, such explanations do not hold. The most likely reason for this is experimental error. Certainly, during the changes between the different pipes, there may well have been unavoidable small alterations in the lip vibration of the artificial mouth. Such changes would be more likely to affect the higher harmonic frequencies of a blown note so making a comparison at these frequencies less accurate. Another source of error may be the clamping arrangement used in the experimental set-up. Although every attempt was made to ensure that the clamping conditions remained constant throughout both the structural response measurements and the artificially blown measurements, it is possible that they changed slightly during the course of the experimentation. With the small vibrational amplitudes being measured, even small differences in the clamping of the pipes would have an adverse effect on the results.

As a final point it is interesting to note that in the measurements presented the change in the magnitude of the wall vibrations is greater between instruments of different material than instruments of different wall thickness. However, it is difficult to draw any conclusion from this, as the geometries of the two groups of instruments were different and the notes produced by the two groups were of different frequencies and loudness.

Chapter 8

Summary and Further Work

8.1 Achievement of Aims

The initial aims of the research, stated in the Introduction (Section 1.4), have been pursued. The following section will restate each of the aims and discuss the extent to which they were fulfilled.

8.1.1 Aim 1

The first aim of the thesis was to determine the main excitation mechanism inducing the structural vibrations in the wall of a brass wind instrument when it is blown.

An experimental set-up was designed to allow the wall vibrations of a simple wind instrument to be ascertained when the brass pipe had been decoupled from both the mouthpiece and then from the air column contained within it. The first decoupling was achieved by separating the pipe and the mouthpiece while the instrument was being sounded and then re-connecting them with a flexible tube of very low stiffness. Thus, the air column standing wave was maintained but the pipe was isolated from any vibrations transmitted directly by the lips via the mouthpiece.

A second experiment saw a pipe of narrower diameter inserted into the brass pipe while the instrument was sounded. One end was fixed directly to the mouthpiece while the other was suspended in a loop of cotton ensuring that no part of the two pipes were in contact. In this case the standing wave of the air column was maintained but completely isolated from the outer brass pipe.

The results clearly show that, while the air column may have a limited effect on the instrument wall, it is the vibration transmitted from the lips via the mouthpiece that is the dominant excitation mechanism. If the vibrational amplitude of the wall does affect the produced sound, the degree to which the tone varies could be due to the individual musicians' playing techniques in terms of the pressure with which they apply their lips to the mouthpiece.

8.1.2 Aim 2

The second aim of the thesis was to determine the effect of both wall material and wall thickness on the vibrational response of a simple lip-reed wind instrument when artificially blown. It was hoped that this would allow insight into the possible influence of either wall material or wall thickness on the tone produced.

The experimental set up was perfected so instruments under test would be measured under exactly the same experimental conditions. In other words, to ensure the excitation force was always applied at the same location with the same force, and the clamping conditions were identical. This allowed an accurate comparison of a particular variable, such as a change in wall material or wall thickness, to be made.

The structural wall resonances were determined for a series of five metal pipes which varied only in wall material and a set of three pipes constructed from the same brass alloy but with different wall thicknesses. Each set of pipes was artificially blown, producing a note of specific pitch, and the induced velocity amplitude variation along the length of the instrument measured and compared to its structural response under the same measurement conditions.

The results reveal that the magnitude of the induced wall vibrations depend on both the instrument's material of construction as well as the thickness of its wall. The differences measured in velocity amplitude can be broadly explained by differences in the structural responses of the instruments at the frequencies constituting the played note.

8.2 Further work

There are a number of different ways in which the study could be extended:

The experiments of Chapter 6 could be extended so that the brass pipe is simultaneously decoupled from both the mouthpiece and the air column. This will prevent both of the hypothesised excitation mechanisms from acting on the walls of the pipe. This should provide confirmation that these are the two excitation mechanisms responsible for exciting the wall vibrations during playing. With the brass pipe decoupled in this way, no wall vibrations should be observed during the playing of the instrument.

The experiments of Chapter 7 could be repeated but this time making sure that the two groups of instruments both have the same internal geometry. From the data measured in these repeated experiments, it should be possible to judge whether it is changes in the wall thickness or in the wall material that have the greatest effect on the amplitude of the wall vibrations (at the frequencies present in the played note).

To investigate to what extent the wall vibrations are communicated to the surrounding air, the Particle Image Velocimetry (PIV) flow visualisation technique could be used to study the air motion close to the pipe wall when an instrument is artificially blown.

To determine whether the wall vibrations really do influence the sound produced, recordings of artificially blown notes could be made for all the instruments (of different wall material and thickness) at some distance from the open end. The frequency spectra of these notes could then be analysed and compared to identify any amplitude variations in the harmonics. Any variation discovered would then be compared with the LDV velocity amplitude measurements for the same pipes.

Finally, the sound recordings could be used in a psychoacoustical investigation. A group of subjects consisting of both musicians and those less 'musically aware' would listen to a set of note pairs and be asked to make judgements on the tonal differences. The set of note pairs would be chosen to allow comparisons to be made between pipes of different wall materials and wall thicknesses. Statistical analysis would then be carried out to determine, for each note pair, whether any variation in tone was detectable.

The psychoacoustical tests could be extended by taking further recordings of the artificially blown notes at the input to the instrument (the player position). In this way, whether players can perceive differences between the tonal quality of the instruments could be investigated. Certainly Smith [1986] reported that it is at the player position that any variation in tone due to change in the wall of the instrument is likely to be detected.

References

J Angster, G Paal, W Garen *et al.*, "The Effect of Wall Vibrations on the Timbre of Organ Pipes," Proceedings of the Acoustical Society of America, JASA,103, 753-754 (1998).

J Backus, "Effect of Wall Material on the Steady-State Tone Quality of Wood Wind Instruments," Journal of the Acoustical Society of America, 36 (10), 1881-1887 (1964).

J Backus, "Wall Vibrations in Organ Pipes and their Effect on Tone," Journal of the Acoustical Society of America, 38, 912 (1965).

J Backus and TC Hundley, "Wall Vibrations in Flue Organ Pipes and their Effect on Tone," Journal of the Acoustical Society of America, 39 (5 (1)), 936-945 (1966).

D Blevins, *Formulas for Natural Frequency and Mode Shape* (Robert Van Nostrand Reinhold, 1979).

CP Boner and RB Newman, "The effect of wall materials on the steady-state acoustic spectrum of flue pipes," Journal of the Acoustical Society of America, 12, 83-89 (1940).

M Campbell and C Greated, The Musician's Guide to Acoustics (Oxford University Press, 1987).

A Cocchi and L Tronchin, "Material and Obsolescence on Flute Quality," Proceedings of the Acoustical Society of America, 103, 763-764 (1998).

JW Coltman, “Effect of Material on Flute Tone Quality,” Journal of the Acoustical Society of America, 49 (2 (2)), 520-523 (1971).

NH Fletcher and TD Rossing, *The Physics of Musical Instruments* (Springer Verlag, New York, 1991).

F Gautier and N Tahani, “Vibroacoustic Behaviour of a Simplified Musical Wind Instrument,” Journal of Sound and Vibration 213 (1), 107-125 (1998).

J Gilbert, S Ponthus, and JF Petiot, “Artificial buzzing lips and brass instruments:experimental results,” Journal of the Acoustical Society of America 104 (3), 1627-1632 (1998).

HP Knauss and WJ Yeager, “Vibration of the Walls of a Cornet,” Journal of the Acoustical Society of America Vol.13, 160-162 (1941).

M Kob, “Influence of Wall Vibrations on the Transient Sound of a Flue Organ Pipe.,” Acta Acustica 86 (4), 642-648 (2000).

B Lawson and W Lawson, “Acoustical Characteristics of Annealed French Horn Bell Flares,” Journal of the Acoustical Society of America 77 (5), 1913-1916 (1985).

W Lottermoser, “Der Einfluß des Materials von Orgel-Metallpfeifen aufihre Tongebung,” Akustische Zeitschrift 3, 63-64 (1938).

DW Martin, “Directivity and acoustic spectra of brass wind instruments,” Journal of the Acoustical Society of America 13, 309-313 (1942).

DC Miller, “The influence of the material of wind instruments on the tone quality,” Science (735), 161-171 (1909).

TR Moore, ET Shirley, and AE Daniels, “Trumpet Bell Vibrations and their Effect on the Sound of the Instrument,” Proceedings of the Stockholm Music Acoustics Conference, SMAC 03, Stockholm, Sweden, Vol.1, 213-215, (2003).

FR Morral, “Metals and Alloys in Musical Instruments,” CIM Bulletin 79 (889), 86-88 (1986).

RW Pyle, “The Effect of Lacquer and Silver Plating on Horn Tone,” Horn Call 11 (2), 26-29 (1981).

RW Pyle, “The Effect of Wall Materials on the Timbre of Brass Instruments,” Proceedings of the Acoustical Society of America, JASA, 103, 753-754 (1998).

A Runnemalm, “Effects of Material Choice and Tooling Methods on Structural Modes of Open Organ Pipes,” Proceedings of the Institute of Acoustics, ISMA'97 19 (5 (2)), 303-308 (1997).

A Runnemalm, L Zipser, and H Franke, “Structural vibration modes of a blown open organ pipe,” Acustica 85, 876-881 (1999).

RA Smith, "Material Vibration and its Influence on Performance of Wind Instruments," Proceedings of the Institute of Acoustics, 317-319 (1981).

R Smith, "The effect of material in brass instruments: a review," Proceedings of the Institute of Acoustics, 8, 81-96 (1986).

PS Watkinson and JM Bowsher, "Vibration Characteristics of Brass Instrument Bells," Journal of Sound and Vibration 85 (1), 1-17 (1982).

G Widholm, R Linortner, W Kausel *et al.*, "Silver, Gold, Platinum-and the sound of the Flute," Proceedings of the International Symposium on Musical Acoustics, Perugia, Italy, Vol.1, 277-280 (2001).

K Wogram, Instrumenenbau 5, 414-418 (1976).

K Wogram, Paper presented at the International Horn Workshop, Los Angeles, 1979.

L Zipser and H Franke, "Vibrations of thin walled tubes and organ flue pipes," Proceedings of SPIE, Second International Conference on Vibration Measurements by Laser Techniques. Ancona, Italy, Vol. 2868, 255-262 (1996).

L Zipser and H Franke, "Vibration Behaviour of Organ Flue Pipes," Proceedings of the Institute of Acoustics, ISMA'97 19 (5), 359-366 (1997).